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AN OPTIMIZATION STUDY OF A LOW THERMAL POTENTIAL POWER SYSTEM.(U)

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AN OPTIMIZATION STUDY OF A LOW THERMAL POTENTIAL POWER SYSTEM

by

J. R. Buckingham
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M. D. Kelleher

September 1976

NAVAL POSTGRADUATE SCHOOL
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THERMOECONOMIC ANALYSIS OF
VAPOR POWER SYSTEMS

A power generating system using the low thermal potential available from the vertical temperature distribution of the ocean is analyzed as a combined engineering and economic mathematical model. The model is optimized for minimum capital cost employing a sequential unconstrained minimization algorithm. Examples of the kinds of engineering and cost information available from the model are presented.

The work reported herein has been supported by the Energy Programs Office, Code L80, of the Civil Engineering Laboratory, Port Hueneme, California; work request N68305 75 WR-S-0068.

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LIST OF SYMBOLS

SYMBOLS	DESCRIPTION	UNITS	
		ENGLISH	METRIC
A	area	in^2	cm^2
c	heat capacity	BTU/lbm-°F	J/kg-°C
C	capacity rate	BTU/hr-°F	kW/°C
d	tube diameter	in	cm
Ds	diameter of the shell	ft	m
f	Fanning friction factor	-	-
F	factor for the tube bundle boiling	-	-
g	inequality constraints	-	-
h	equality constraints	-	-
h	heat transfer coefficient	BTU/ft^2-hr-°F	kW/m^2-°C
H	enthalpy	BTU/lbm	J/kg
H'	enthalpy corrected for subcooling	BTU/lbm	J/kg
hs	height from midplane of the boiler to top of the tube bank	ft	m
i	vapor quality	-	-
k	thermal conductivity	BTU/hr-ft-°F	kW/m-°C
Kc	pressure loss factor at the tube entrance	-	-
Ke	pressure loss factor at the tube exit	-	-
L	tube length	ft	m
M	mass flow rate	lbm/hr	kg/hr
n	average number of tubes in a column in the condenser	-	-
Nsq	specific speed of a pump	RPM	RPM
NTU	number of heat transfer units of an exchanger	-	-
Nu	Nusselt number	-	-
p	pressure	lbf/in^2	kp

P	power	BTU/hr	MW
pi	tube pitch	in	cm
Pr	Prandtl number	-	-
q	heat flux	BTU/hr-ft²	kW/m²
Q	heat transfer rate	BTU/hr	MW
Rn	heat transfer resistance (R1,R2,R3,R4)	hr-°F/BTU	°C/kW
Re	Reynolds number	-	-
s	entropy	BTU/lbm-°R	kW-hr/kg-°K
S	number of shells	-	-
SA	segmental area	ft²	m²
t	tube wall thickness	in	cm
T	temperature	°F	°C
u	velocity	ft/sec	m/sec
U	overall heat transfer coefficient	BTU/ft²-hr-°F	kW/m²-°C
v	specific volume	ft³/lbm	m³/kg
VL	vapor load	lbm/hr-ft³	kg/hr-m³
y	proportionality factor in the transition region	-	-
$	capital cost	dollars	dollars

GREEK SYMBOLS

ϵ	heat transfer effectiveness	-	-
η	efficiency	-	-
λ	proportionality factor in the transition region	-	-
μ	dynamic viscosity	lbm/sec-ft	N/sec-m
ν	kinematic viscosity	ft²/sec	m²/sec
ρ	density	lbm/ft³	kg/m³
σ	surface tension	lbf/ft	N/m
ϕ	free flow/frontal area ratio	-	-

SUBSCRIPTS

ab	absorbed
b	boiler
bulk	average of the inlet and outlet temperatures
c	condenser
c	critical pressure
c	the colder fluid in the heat exchanger
car	carnot efficiency
cp	circulation pump
cyc	thermodynamic cycle
f	saturated liquid
fg	change in the quantity as the fluid changes from all liquid to all vapor
fp	feed pump
g	saturated vapor
h	the hotter fluid in the heat exchanger
he	heat exchanger
i	inside
in	fluid entering the exchanger
l	laminar flow regime
m	mean temperature difference
o	max. possible temperature difference
o	outside surface
out	fluid leaving the exchanger
p	at constant pressure
rej	rejected
sw	seawater
sys	the system
Tr	transition regime
Tu	turbulent regime

tur	turbine
w	wall
wf	working fluid
C	maximum temperature difference
1	state point 1
2	" " 2
2s	" " 2s
3	" " 3
4	" " 4
5	" " 5
5s	" " 5s

I. INTRODUCTION

A. BACKGROUND

As of 1974, the United States contained only 6% of the world population but was using 33% of the energy consumed each year according to Ref. 1. Approximately one third of the oil used in the U.S. is imported. This heavy dependence on foreign sources for energy coupled with the exhaustion of domestic fossile fuels, particularly oil, in the foreseeable future is causing the U.S. military and civilian sectors of the economy to search for energy sources that would be inexhaustible and independent of foreign control. Increasing fuel costs and environmental problems are other forces pushing the search for alternatives. Nuclear power, once considered the cure-all, is facing severe problems. Costs of construction, operation, and fuel have risen dramatically. Questions concerning the safety of operation of the nuclear plants and of the storage of nuclear wastes have become political issues that have slowed the exploitation of nuclear power. The present nuclear reactors use a fuel that, like petroleum, has a limited availability and whose price is increasing rapidly. The breeder reactor, which would create more fuel than it burns, and the fusion reactor, which would use the hydrogen found in water, are still a long way from producing power for a world whose demands for energy increase each year.

Many people advocate the exploitation of the so-called "free" energy sources. These sources are: (1) the kinetic

energy of moving fluids, such as winds and ocean currents; (2) the potential energy of tides and rivers; (3) the heat generated inside the earth; (4) the direct conversion of solar energy into electricity and heat; and (5) the use of sun warmed water in conjunction with a source of colder water to provide the temperature differential to run a vapor power cycle. None of these sources are truly "free". What is happening is the trading off of the transportation, processing, and environmental costs of a concentrated energy source for the capital, operating, and social costs of converting a very diffuse source into a more concentrated form such as electricity.

The earth can be thought of as a giant heat engine, absorbing energy from the sun and reflecting energy as thermal radiation back to space as pictured in Ref. 2. Since the poles receive less energy than the equator, the atmosphere and ocean attempt to distribute the energy more evenly over the earth. The air is heated by absorption of sunlight and by contact with the surface of the earth. The heated air rises thus setting up surface wind currents as cooler air tries to replace the rising air. The shear stress created by the relative motion between the air and the surface water causes the water to move thus creating the surface water currents. Many currents in the oceans, regardless of depth, originate with the wind shear stress at the air-surface interface. The circulation patterns of the oceans become extremely complex due to the influence of the rotation of the earth and the shapes of the ocean basins.

In general the surface waters near the equator are warmed by the sun and flow towards the poles giving up energy by evaporation, and by convection to the air and by radiation back to space. As these currents cool, some of the water becomes dense enough to sink and then flows back to the equator along the bottom.

The relationship between salinity and temperature and the density of seawater is such as to create a stable stratification of the ocean in many areas. A salinity decrease or a temperature increase reduces the density. The total effect is to create a warm surface current overlaying the cold water returning from the poles. In the tropical waters the surface temperature may exceed 80°F (26.7°C) while 3,000 to 6,000 ft. (914 to 1828 m) down the temperature may be about 40°F (4.4°C). This temperature difference can be used to run a man-made heat engine, like the Ocean Thermal Energy Converion (OTEC) power plant concept.

The proper choice of a site is critical to the cost of such a heat engine. Since the temperature difference is low compared to conventional thermal power generating methods which typically use temperature differentials exceeding 500°F (260°C), the effect of a 1°F (0.56°C) loss in the ΔT is much more costly at the low thermal differentials, which are usually less than 50°F (27.8°C) in the ocean. Therefore, the vertical temperature profile of the oceans will control the location of an OTEC power plant. The surface temperatures vary with location and with time. At most places in the ocean there is a seasonal change of the surface water temperature of 5 to 8°F (2.8 to 4.4°C) except in the tropical areas that remain relatively constant according to Ref. 3. However, all regions of the oceans are affected by wind-generated mixing of the surface water that changes the surface temperature by a degree or so and a daily cycle of a few tenths of a degree. Some of the possible U.S. near-shore sites being considered are the waters around the Hawaian Islands and the Gulf Stream off the southeastern U.S.

In the past, the engineering analysis and the cost estimation have been conducted as if they were separate

functions. In a system with a small temperature differential, such as this one, the engineering design and economic analysis should be linked together as a single procedure because of the capital intensive nature of the ocean thermal power plant. For example, the heat transfer equation ($Q = UA\,\Delta T$) is coupled to the cost equation for heat exchangers ($\$ = K\,A^n$) through the surface area of the tubes. Because of the small ΔT involved the area must be very large to supply the same Q. The other components are all linked together by the engineering and cost equations so that the problem cannot logically be broken into an equipment by equipment optimization. The present analysis considers a closed vapor power system operating on the thermal potential available from the ocean. The economic optimization of the system is carried out to show what information can be gotten, what conclusions can be reached from such analysis and where research effort should be expended to improve the design of the system. This study does not attempt to predict the cost of such a system or to make specific recommendations about the parameters.

A schematic diagram of the system is shown in Figure 1. The warm surface water, the Gulf Stream for example, is pumped through a heat exchanger. There some energy is transferred by boiling the working fluid such as ammonia. The working fluid is piped to a turbine where some of the energy is converted to mechanical energy then to electrical energy in a generator. The unavailable portion of the absorbed energy is rejected to the environment in the condenser and the working fluid is returned to the liquid state. The low temperature in the condenser is maintained by pumping cold ocean water up from the depths. Finally the working fluid is pumped back to the boiler.

The Ocean Thermal Energy Conversion program has received much attention in the last several years from many

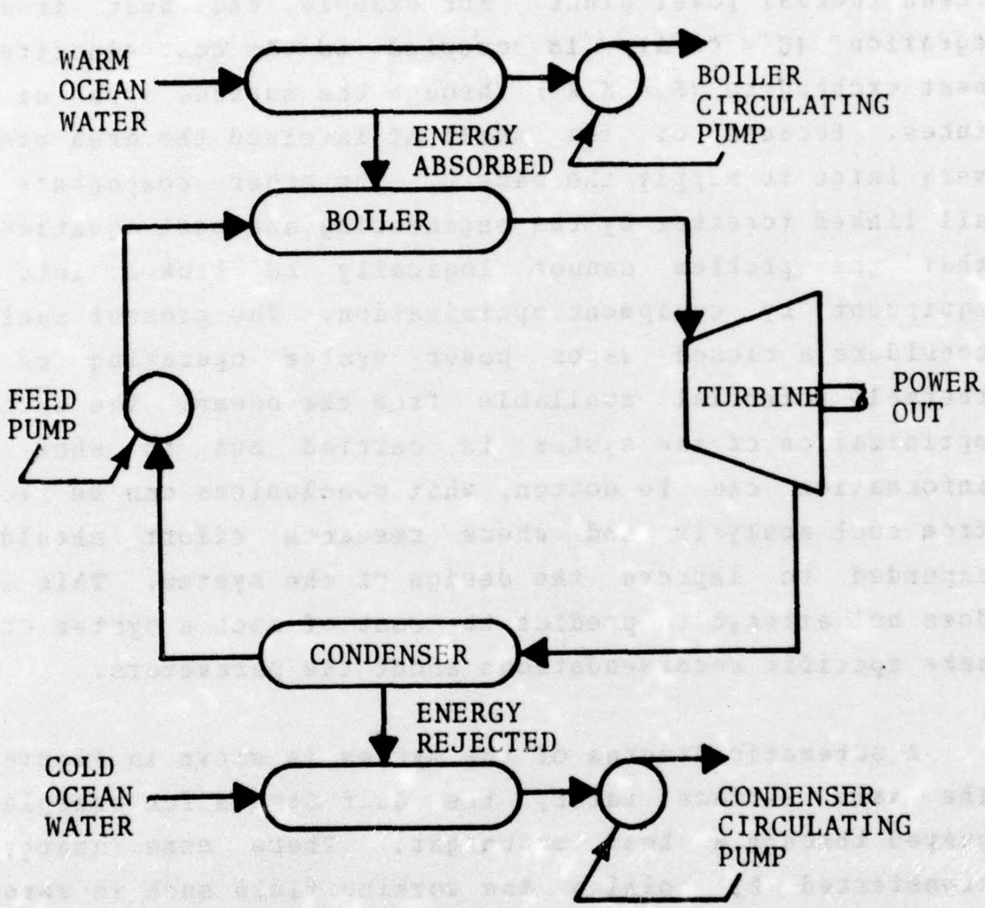


FIGURE 1. SYSTEM SCHEMATIC DIAGRAM

researchers funded at first primarily by the National Science Foundation (NSF). Since the establishment of the Energy Research and Development Administration (ERDA), new research contracts are funded mainly by this agency. The U.S. Navy is also funding some research in the OTEC area. Ref. 4, 5 and 6 give a good summary of the work that has been done. While much of the work has concentrated on the heat exchangers, the other components and the system as a whole are receiving attention. Some researchers are concentrating on the environmental and legal questions of such systems. Other teams are considering how to use the energy supplied from an OTEC power plant.

In any design, trade-offs must be made. Capital cost can be traded for operational costs. Pumping power can be traded for heat exchanger size. Trade-offs can involve items that have a common measure such as dollars; some other trade-offs are very hard to measure in common terms such as social and environmental costs. The trade-offs may involve only one component such as a pump or a turbine, but most often the trade-offs affect more than one component or possibly the entire system.

The trade-off process should continue until an optimum design is obtained. From Ref. 7, "The aim of optimization is the selection, out of the multiplicity of potential solutions, of that solution which is the best with respect to some well-defined criterion." In order for the plant design to be optimum, the components must be optimized within the context of the entire system. Some of the parameters for a piece of equipment may have no direct effect on the optimum design of another portion of the system, but some parameters will have significant effects. For example, a heat exchanger is made up of a large number of tubes of a given diameter and length and is designed to have a certain heat transfer rate for a given flow rate of

water pumped through the tubes. The diameters, length and the number of tubes may be optimized for lowest cost of the heat exchanger. This approach leaves out the effect that the tube diameter and the tube length have on the pressure drop across the heat exchanger. Minimizing the cost of the heat exchanger involves minimizing the surface area. This may lead to increasing the pressure drop across the exchanger. The higher the pressure drop the more powerful and the more expensive the pump must be to maintain the flow velocity. This is just one example of the hundreds of trade-offs possible.

Unfortunately, what happens in many cases is that each group of experts working on a particular component optimize that item around some given conditions that may have been set by teams working on some other section of the system. Little analysis is made of how the entire system responds to the coupling constraints. All that the pump manufacturer wants is that the buyer tell him what the flow rate, head and service conditions are and he will quote a price. Before the buyer can decide on the flow rate and head he should first know how the cost of the pump varies as flow rate and head vary so that those two parameters can be optimized in the context of the system.

The problem then becomes one of acquiring sufficient information on how costs vary as certain parameters change. Each manufacturer is able to make estimates of costs for his product but he is reluctant to devulge his information to buyers or competitors. This makes constructing cost curves a hazardous process at best, since the data must be gathered by other methods that may cause large unexplained variances.

Sometimes the optimization analysis is made with assumptions that make the answers of questionable value. For example, sometimes the heat transfer coefficient, U, is

assumed fixed in the equation $q = UA\,\Delta T$. But, as shown in Ref. 8, for boiling the heat transfer coefficient is very sensitive to the temperature difference with the result that $q = UA'\Delta T^{3.33}$. The The accuracy of the engineering equations should be kept in mind. Most of the equations are correlations to fit experimantal data. The correlation was develcped for some particular set of data and may disagree significantly when compared to data taken by other experimenters. The limitations of the various relations must be kept in mind when reviewing the results. For a system the size of the OTEC plant, a pilot plant should be constructed in order to prove out the answers given by the calculations before expensive mistakes are made.

As a design progresses, the analysis begins with a general picture containing many simplifying assumptions and proceeds foreward with more details added at each stage. This report is the second one written at the Naval Postraduate School on the subject of OTEC. Commander Furman Sheppard, USN, in his analysis, Ref. 9, attempted to show what kinds of information could be gained from a combined thermal economic model. He advocated the use of zone analysis, as develcped by Wismer in Ref. 10, for large complex systems. In Wismer's approach to zone analysis, the system is broken down into zones containing one or more components. The zones are connected by linking variables. Two methods of optimization are possible. For the first method, in each zone, the zone parameters are optimized while the linking variables are held fixed. Then, the zone parameters are held fixed while the linking variables are optimized. The process iterates between optimizing the zone parameters and the linking variables until convergence criteria are satisfied. In the second method the zones are cut off from each other and each linking variable is separated into two variables, one on each side of the cutting plane. The optimization proceeds until the linking

variables on each side of the cuts are equal.

Sheppard's system did not produce vapor in a boiler but, instead, maintained the working fluid in the liquid state in the warm seawater heat exchanger and vaporized the working fluid, ammonia in his model, by expanding it through a "black box" called a vaporizer. His optimization analysis was confined to the zone containing the warm seawater exchanger, the feed pump, and the warm water circulating pump.

E. OBJECTIVES

The present analysis improves the previous model analysis by replacing the liquid-to-liquid heat exchanger and the vaporizer with a boiler. The boiler model is to be of sufficient detail so that a realistic design is represented. The other major components of the system, such as the turbine, the condenser, and the condenser circulating pumps, are included. A nonlinear programming technique is used to perform the thermal-economic optimization of the completed system. The research reported here is intended to show the usefulness of optimization analysis in the design of a low thermal difference system and to demonstrate some of the problems that must be considered in this type of analysis.

II. THE MODEL

A. GENERAL DESCRIPTION

Vapor power generating cycles usually consist of four processes. Energy is absorbed from a high temperature source and is used to vaporize a working fluid in a boiler. Next, work is extracted by expanding the vapor in an engine, exhausting the vapor at a lower temperature and pressure. The working fluid is then condensed back to the liquid state by rejecting energy to the lower temperature energy sink in the third process. Finally, the liquid is returned to the higher pressure of the boiler by a pump. In the following analysis, the energy source is a current of ocean water heated by the sun and the energy sink is assumed to be colder ocean water from deep currents. The working fluid is assumed to be propane although many other fluids are possible, such as ammonia or the Freons.

Six components are modeled in the following analysis. The hot side heat exchangers, the boilers, vaporize the working fluid which is expanded through a turbine exhausting to a condenser that rejects the unused heat to the cold sink. Three pumps are included, a feed pump returns the working fluid to the boiler and two pumps circulate the warm and cold waters through the boiler and the condenser, respectively. The arrangement of the system is shown in Fig. 2. The nodes are numbered to correspond with the fluid state points shown later, in Fig. 3.

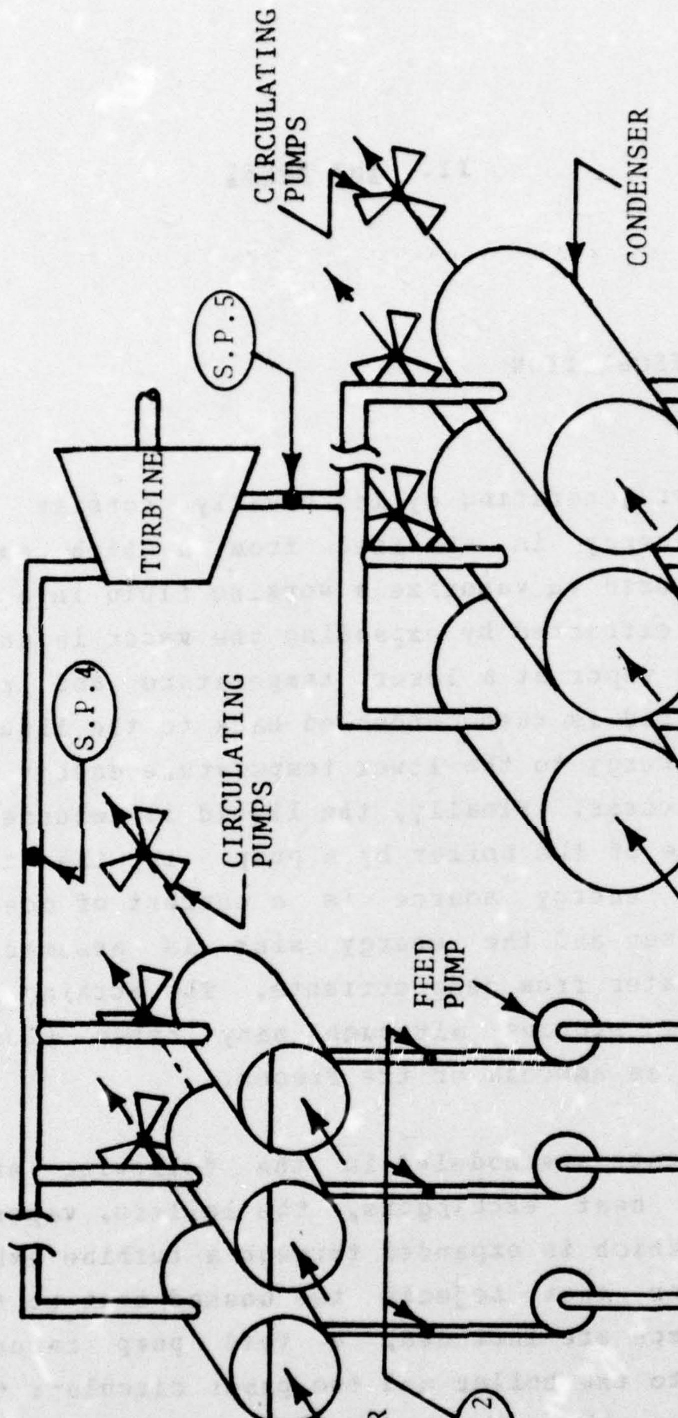


FIGURE 2. SYSTEM ARRANGEMENT

The analytical model consists of the equations describing the system and a set of parameters to be optimized so as to minimize the capital cost of the plant subject to certain constraints. The objective function, the function to be minimized, is the sum of the capital costs of the pumps, the boiler and the condenser. Although the turbine is a significant capital cost and would affect the design of the system, it is not included in this model. There are two classes of constraints. The first class is the explicit constraints that specify the upper and lower bounds on the parameters. The other class of constraints is the implicit constraints. These are the engineering equations that describe the operation of the power plant. For example, a constraint might say that the temperature inside the boiler must be less than the temperature of the warm seawater because the equations fail if the requirement is not satisfied. Another example would be a power balance that requires that the sum of the energy flows into and out of the system be zero. Additionally, they proscribe the region that the optimization routine may search within for the minimum of the problem. The implicit constraints are of two types, equality and inequality constraints. Inequality constraints set one-sided bounds on the feasible region. Some of the inequality constraints are restrictions placed on the maximum or minimum values of the parameters due to judgements made on the basis of practical engineering considerations. The equality constraints are relationships that must be strictly satisfied at the optimimum. The system of equations describing the model fit into the following form:

$$\text{minimize} \quad f(X)$$

$$\text{subject to:} \quad g_i(X) \geq 0, \; i=1,2,\ldots,m$$

$$h_i(X) = 0, \; i=m+1,\ldots,m+n$$

where X is the vector of the parameters to be optimized, $f(X)$ is the capital cost function, $g_i(X)$ are the inequality constraints and $h_i(X)$ are the equality constraints. The parameters are the dimensions of the boilers and the condensers, the boiling and condensing pressures, the sea water velocities, and the mass flow rate of the working fluid through the system.

The basis for the economic framework must be set out at the start. All measurable costs are relevant to the final design of the power plant, but, for some types of analysis and design, many costs can be ignored. The life cycle cost of the CTEC must be considered before the decision is made to commit the U.S. and the rest of the world to the consequences of choosing this energy generation scheme as even a partial solution to the energy problem. The life cycle cost consists of the capital costs, the financing costs, and the operating costs. The trade-off of operating cost and capital cost is not considered in this research, because the operating cost is, for the most part, independent of the overall dimensional characteristics of the components. Operating costs are dominated by the structure, site location, and maintenance requirements. Such things as the kind of instrumentation, controls, bearings, and structural materials have more effect on the operational costs than the length of the boiler or the diameter of the boiler shell. Therefore, only the capital costs of the major components, except the turbine, are considered.

To be useful for model analysis cost data must be transformed into a cost estimating relationship (CER) that shows how cost varies with some variable or some combination of variables. The costs must be demonstrated to be highly correlated with the variable chosen. Since the cost relation is only a correlation, the bounds on the accuracy of the equation should be specified by those who develop it. Since the relation is arrived at from a finite body of data, the CER is directly applicable only in the range of the data collected. Extrapolation of the CER outside the range is dangerous because there is no data in that region to support the assumed curve. However, the value of CER's is the use of them to enable the estimator to make a better analysis of a new design than he could if the information is not available. The relevant range of the CER must be kept in mind when analyzing the solutions from the standpoint of the reliability of the answers. Another factor to consider when using cost estimating relationships is whether the equipment the data was taken from is comparable to the system the analyst is considering. If the design, the technology, or the application of the proposed equipment is significantly different from that from which the data is taken, the CER may give unreliable answers.

This analysis begins with the development of the engineering equations that constrain the system. This is followed by the development of a cost framework based on capital costs as the function to be minimized. The engineering and economic relations are linked together in the objective function and the constraints. A description of the algorithm that performs the optimization is included.

B. CYCLE ANALYSIS

The building of the model begins with the analysis of the thermodynamic cycle. Figure 3 is a temperature-entropy diagram of the working fluid on which the cycle is shown. These numbered nodes represent the thermodynamic state points at that location in the system and correspond to the numbers in Fig. 2. The boiling and condensing processes are assumed to be reversible processes. In the pump and the turbine the reversible process is indicated by the subscript "s" on the state point number. In the actual system there are irreversible losses inside each component that are left out of the analytical model or lumped in with other losses. The pressure drops due to frictional losses in the boiler and the condenser are neglected.

State point 1 assumes that the liquid leaving the boiler is saturated. State points 2s and 2 represent the conditions leaving the feed pump. If there were no losses in the pump, the fluid would be at 2s. The fluid enters the boiler to be heated, first, to saturation conditions at state point 3 then boiled at essentially constant pressure. State point 4 shows the vapor leaving the boiler as a saturated vapor. The lines from point 4 to 5s and 5 represent the expansion through the turbine. The path to state point 5 accounts for the losses due to friction and other irreversible processes in the turbine. The remaining heat is then rejected from point 5 to point 1, again assumed to be a constant pressure process.

The theoretical heat transfer rate in the boiler and the condenser is found by subtracting the enthalpy at state points 2 and 1 from the enthalpy of state points 4 and 5, respectively, and multiplying by the total mass flow rate of the working fluid.

$$Q_{ab} = M_{wf}(H_4 - H_2) \qquad (1)$$

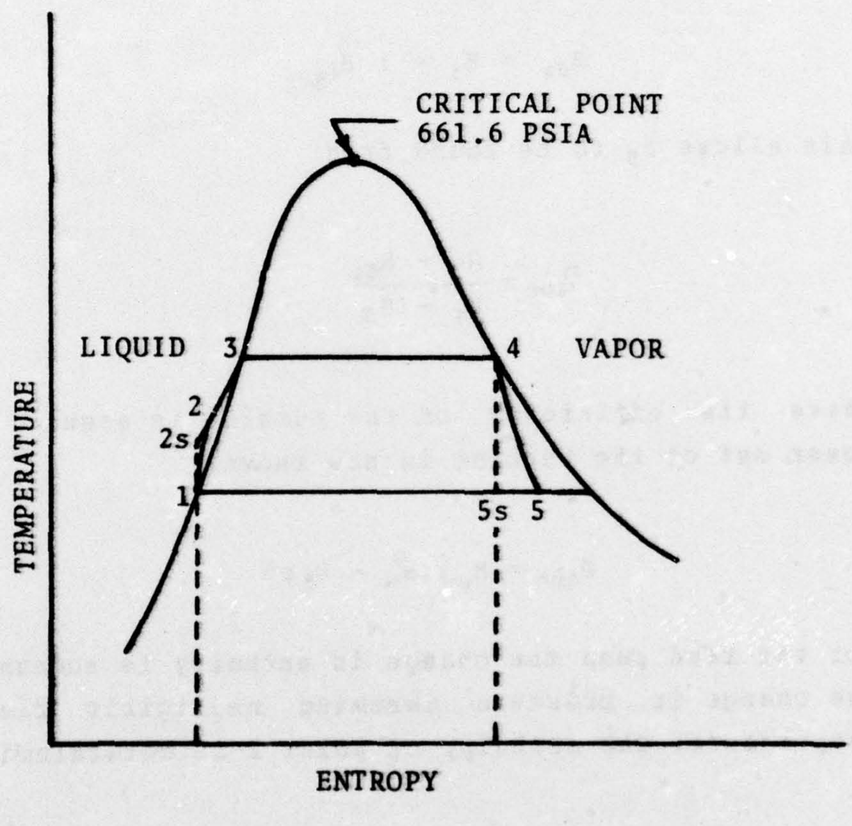


FIGURE 3. THERMODYNAMIC CYCLE

$$Q_{rej} = M_{wf}(H_5 - H_1) \qquad (2)$$

The enthalpy at state point 5 is determined by first finding the moisture content of the fluid exiting the turbine.

$$i = \frac{s_4 - s_1}{s_{fg}} \qquad (3)$$

The enthalpy at point 5s is determined from

$$H_{5s} = H_1 + i\, H_{fg} \qquad (4)$$

This allows H_5 to be found from

$$\eta_{tur} = \frac{H_4 - H_{5s}}{H_4 - H_5} \qquad (5)$$

where the efficiency of the turbine is assumed known. The power out of the turbine is now known.

$$P_{tur} = M_{wf}(H_4 - H_5) \qquad (6)$$

For the feed pump the change in enthalpy is accounted for by the change in pressure assuming negligible rise in the temperature. The enthalpy of point 2 is determined from

$$\eta_{fp} = \frac{v_1 (P_{2s} - P_1)}{H_2 - H_1} \qquad (7)$$

where the efficiency is assumed known. The power required to drive the pump is determined from

$$P_{fp} = M_{wf}(H_2 - H_1) \quad (8)$$

To gain a better feeling for the low thermal potential systems and to make comparisons with other systems several efficiencies are worthwhile calculating. The first is the Carnot efficiency which defines the maximum efficiency possible for a system operating between two temperature limits in degrees absolute, °R (°K).

$$\eta_{car} = 1 - \frac{T_{cold}}{T_{hot}} \quad (9)$$

The next efficiency calculation takes into account the theoretical thermodynamic cycle and is the ratio of the net power out of the cycle to the heat taken up by the cycle

$$\eta_{cyc} = \frac{P_{tur} - P_{fp}}{Q_{ab}} \quad (10)$$

This accounts for the thermodynamic inefficiencies inherent in the cycle. Finally, the system efficiency takes into account the other internal consumers of power: the circulation pumps, the pipe losses, and any other losses, such as the antifouling equipment. This calculation accounts for all of the mechanical and electrical inefficiencies of the system.

$$\eta_{sys} = \frac{\text{NET POWER AT THE BUS BAR}}{Q_{ab}} \quad (11)$$

For example, if the hot water is at 75°F (23.9°C) and

the cold water is assumed at 40°F (4.4°C) the Carnot efficiency is 6.5%. For one typical set of assumptions, as shown in Table 5, where the turbine and the pumps are assumed to be 85% efficient, the cycle efficiency is 2.6% and the system efficiency is 2.4% for one feasible design.

C. ENGINEERING FRAMEWORK

This section describes the design of the components that make up the system. The level of detail of the analysis provides a physically realistic model and closes the loop. The pipe friction losses on the cold and hot water inlets as well as the pipe losses in the working fluid side are neglected. The turbine is included only as an energy transducer and no attempt is made to obtain costs, physical dimensions, or fluid flow characteristics. The efficiencies of the turbine and the pumps are assumed as known constants. The heat absorbed in heating the working fluid to the saturated state is a small portion of the total energy taken up in the boiler; therefore, the sensible heating of the fluid is neglected.

1. The Boiler

The boiler consists of a number of tube-in-shell heat exchangers connected in parallel as shown in Fig. 2. Figure 4 shows the internal arrangement of one shell. The tube bundle is horizontal and boiling occurs on the outside surface of the tubes. The tube bundle is submerged in the working fluid. The working fluid is maintained at a level to just cover the top row of tubes. The warm seawater is pumped through the inside of the tubes.

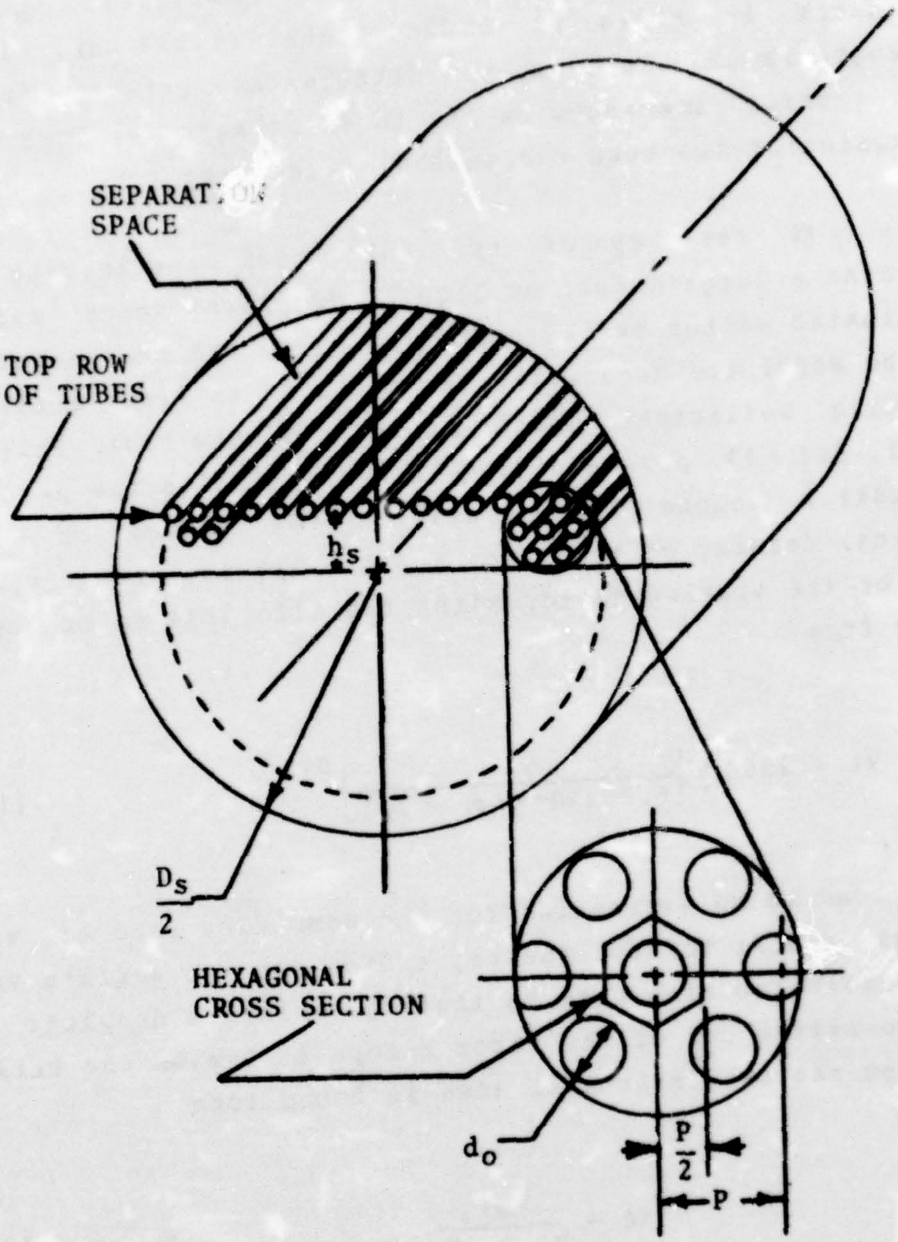


FIGURE 4. BOILER CROSS SECTION

The clearance between the tubes is fixed at 0.5 inches (1.27 cm) to allow for flaring of the tube ends or other space and strength requirements. Similarly, the tube thickness is fixed at 0.028 inches (0.071 cm), since no strength requirements due to differences between internal and external pressures on the tubes or material loss due to corrosion of the tube surfaces are accounted for.

At the top of the tube bank the leaving vapor contains a large amount of liquid droplets which must be eliminated either in the boiler or by an external separator. In the model the separation takes place in the boiler by allowing sufficient space above the tube bank. Palen and Small, Ref. 11, presented a method for finding the necessary separation volume that considers the effects of surface tension, density of the vapor and the liquid, and the mass flow of the working fluid. First the allowable vapor load is found from

$$VL = 2290\rho_f\left[\frac{\sigma}{6.86\times10^{-5}(\rho_f - \rho_g)}\right]^{0.5} \quad (12)$$

Vapor load is an expression for the mass flow rate of vapor through a unit volume. The expression for the maximum vapor load limits the flow rate so that the moisture droplets are able to settle out of the vapor before it leaves the boiler. Then the required segmental area is found from

$$SA = \frac{M_{wf}}{VL\ L\ S} \quad (13)$$

The segmental area is the cross sectional area from the top of the tube bank to the top of the shell. Knowing the segmental area the height of the tube bank above the main

diameter is found from

$$SA = 0.25\ Ds^2\ \mathrm{ARCOS}\left(\frac{2\ hs}{Ds}\right) - hs(0.25\ Ds^2 - hs^2)^{0.5} \qquad (14)$$

Palen and Small also present a much simpler method of determining the total height of the tube bank. The height of the separation volume is set at 40% of shell diameter which results in "hs" being equal to 0.1 Ds. Either method can be used alone or the first method can be used as a constraint on the second. For this problem the segmental area is found by both methods with the first used as a check on the values given by the second.

Since the heat flux calculation is based on a single tube, the number of tubes per shell must be determined in order to find the total energy absorbed by the heat exchanger. The tubes are assumed to be laid out in a 60 degree triangular pattern for maximum compactness. The tubes are located at the vertices of an equilateral triangle. Two cross sectional areas are calculated. The area of a hexagon whose minor radius is equal to one half the pitch is determined from

$$A = 6\left(\frac{P}{2}\right)^2 \mathrm{TAN}\ 30° \qquad (15)$$

Then the effective cross sectional area of the tube bank is found by subtracting the clearance area, SA, from the cross sectional area of the shell after correcting the shell diameter for a clearance space to allow for free flow channels both within the tube bundle and around the outside of the tube bundle. The effective cross sectional area of the tube bank is divided by the area of the hexagon, found

above, to determine the number of tubes in a shell.

The flow of the liquid and vapor through the tube bank is assumed to be turbulent. The incoming subcooled working fluid from the feed pump mixes with the recirculating fluid returning from higher up in the tube bank. The circulation of the fluid is induced by the boiling process. This mixture enters the tube bank and begins the boiling process. Near the bottom of the tube bank the heat transfer goes mostly to bring the fluid up to the boiling temperature. After the fluid reaches saturated liquid conditions, stable boiling begins. The vapor bubbles rising through the surrounding liquid increase the turbulence around the tubes. As the liquid continues up through the tube bank, the local boiling temperature is decreasing due to the decreasing hydrostatic head. This decrease in the boiling temperature increases the temperature difference and increases the heat flow and increases the bubble generation rate. The vapor generation may become so rapid that a condition called vapor blanketing may occur in the interior of the tube bank where the liquid cannot flow inward fast enough to displace the vapor flowing through the region. This is similar to burnout in convective boiling in that the tubes dry out and the heat transfer rate decreases. This effect becomes pronounced only near the critical heat flux according to Starczewski, in Ref. 8.

All of the heat and mass flow variations in the tube bundle are averaged by performing the calculations for a single tube located at the mid height of the tube bank. This tube is assumed to represent the average of all the tubes in the tube bank. The boiling situation is assumed to be saturated pool boiling. Quoting from Collier, Ref 12, "pool boiling is defined as boiling from a heated surface submerged in a large volume of stagnant liquid". Since the

fluid volume seen by the tube is neither large nor stagnant, a correction is needed for the effect of being in a tube bundle. Saturated boiling is defined as boiling at a constant temperature and pressure as the liquid changes to a vapor. This is probably not strictly true because the boiling temperature is controlled by the local pressure at each tube, and the fluid is not reaching a stable temperature as it rises in the bundle. The situation is probably closer to convective boiling since the fluid is boiling in a confined channel between the tubes. An additional effect in the boiler is the fact that the hydrostatic head at the bottom of the tube bundle causes the boiling to take place at a higher pressure at the bottom than the boiling at the top of the bundle where the pressure is equal to the boiler exit pressure. This effect is accounted for by using the average head of the bundle in the heat transfer equations.

The heat transfer analysis of the boiler begins with the basic equation

$$Q = UA\,\Delta T_m \tag{16}$$

where U is the overall heat transfer coefficient, A is the area through which the heat is transferred and ΔT_m is a properly defined mean temperature difference. In this paper, the product UA is treated as a single quantity called the thermal conductance. One may look at the equation as an analogue of the basis electrical equation I = E/R where I is equivalent to Q, E is the same as ΔT_m and UA is 1/R. This analogy is used when developing the expression for UA.

In order to account for the effects of the tube seeing many nearby neighbors, the equation is modified by the inclusion of a factor, F, to account for the effects of

the surrounding tubes. Equation 16 is changed to

$$Q = UA\ F\ \Delta T_m \qquad (17)$$

where F is defined as the ratio of the heat flux of the tube bundle to the heat flux for a single tube in a stagnant pool boiling situation. According to Ref. 13, a report published by workers at Heat Transfer Research, Inc. (HTRI) of Alhambra, California, F is greater than one. In Figure 5, which is sketched from a figure in their report, F ranges from 4.73 at ΔT_m = 8°F to 2.63 at ΔT_m = 40°F. Since no data is given on the conditions of the test, F is assumed to be fixed at three.

In order to find ΔT_m, the heat exchanger effectiveness, ε, is determined. The effectiveness is defined as the ratio of the actual heat transfer rate to the maximum possible heat transfer rate as shown by

$$= \frac{C_h\ (T_{h,in} - T_{h,out})}{C_{min}\ (T_{h,in} - T_{c,in})} \qquad (18)$$

where

$$C = (M\ c_p)_{fluid} \qquad (19)$$

$$C_{min} = \text{minimum of } C_h \text{ or } C_c \qquad (20)$$

In the boiler, $T_{h,in}$ is the temperature of the entering seawater, and $T_{c,in}$ is the saturation temperature of the working fluid at the pressure at the mid height of the boiler. Reference 14, by Kays and London, contains equations for finding the effectiveness for various flow configurations. In addition to finding C_{min}, C_{max} must be

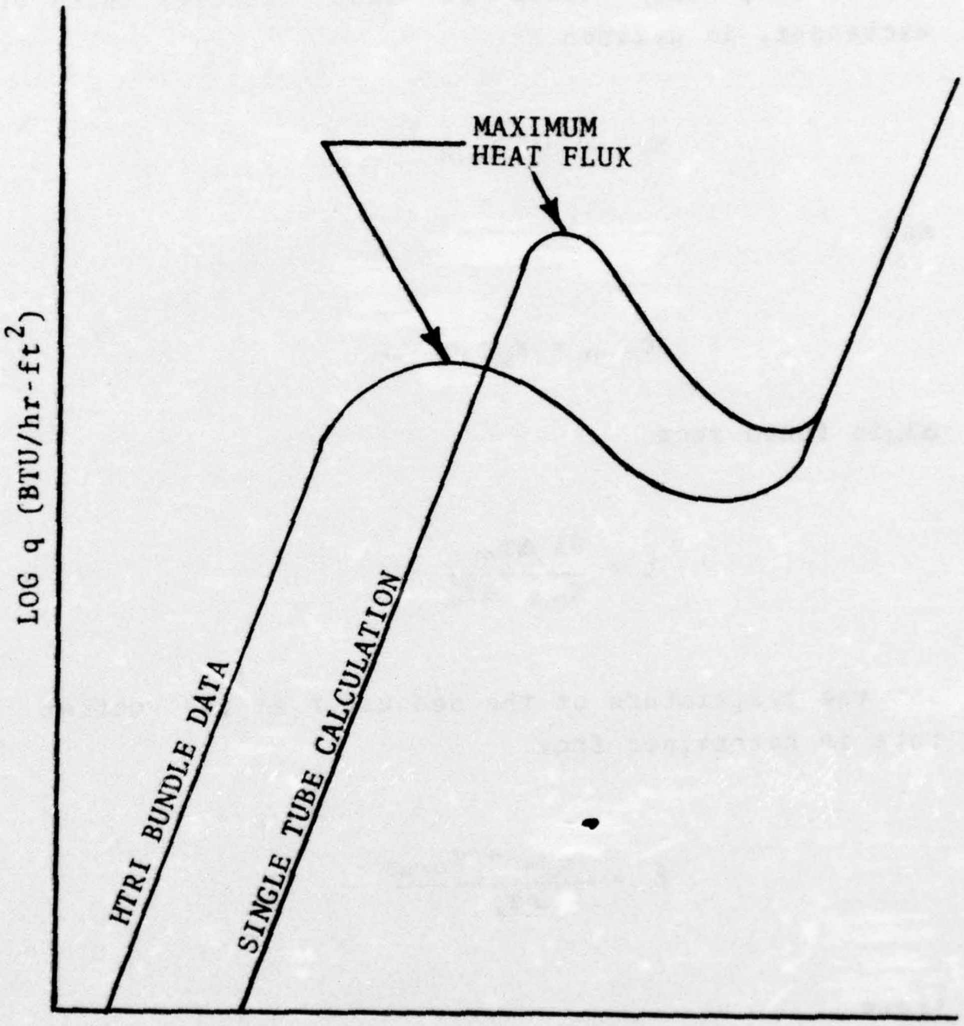


FIGURE 5. COMPARISON OF A SINGLE TUBE VERSUS A TUBE IN A BUNDLE

known in order to decide which equation applies. For the case of boiling, C_{max} is infinite; therefore, regardless of the flow pattern

$$\varepsilon = 1 - EXP(-NTU) \tag{21}$$

where NTU, the number of heat transfer units of a heat exchanger, is defined as

$$NTU = UA/C_{min} \tag{22}$$

and

$$C_{min} = M_{sw}\ c_{p,sw} \tag{23}$$

ΔT_m is found from

$$\varepsilon = \frac{UA\ \Delta T_m}{C_{min}\ \Delta T_o} \tag{24}$$

and the temperature of the sea water at the outlet of the tube is determined from

$$\varepsilon = \frac{T_{h,in} - T_{h,out}}{\Delta T_o} \tag{25}$$

where

$$\Delta T_o = T_{h,in} - T_{c,in} \tag{26}$$

The quantity UA is determined next. Figure 6

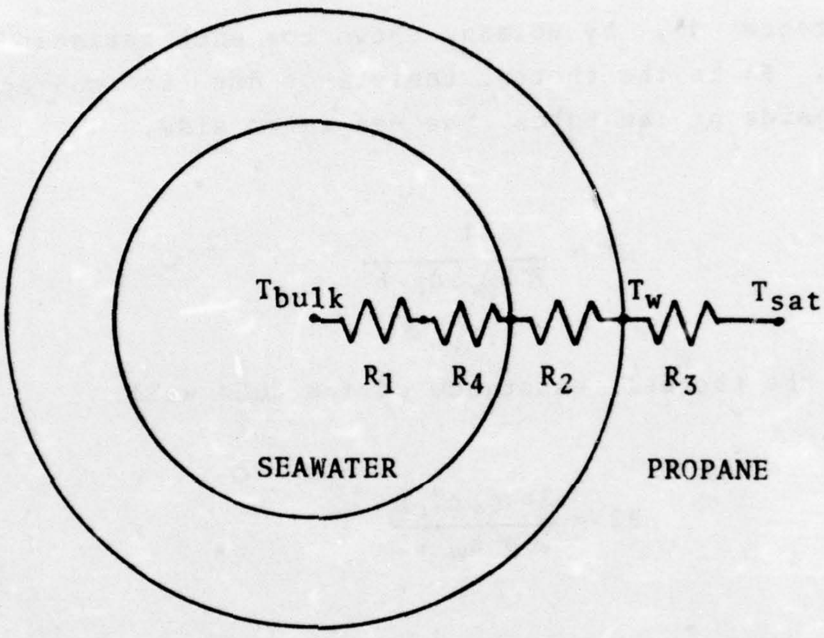


FIGURE 6. THERMAL NETWORK FOR A SINGLE TUBE

illustrates the method used. If UA is pictured as the reciprocal of resistance then the flow of energy from one fluid to the other can be pictured as a series electrical circuit and UA is determined from

$$UA = 1/(R1 + R2 + R3 + R4) \qquad (27)$$

Reference 15, by Holman, shows how each resistance can be found. R1 is the thermal resistance due to convection on the inside of the tubes, the sea water side.

$$R1 = \frac{1}{\pi h_{sw} \; d_i \; L} \qquad (28)$$

R2 is the thermal resistance of the tube wall.

$$R2 = \frac{\ln(d_o/d_i)}{2 \pi k_w \; L} \qquad (29)$$

R3 is the thermal resistance attributed to boiling on the outside of the tubes.

$$R3 = \frac{1}{\pi h_{wf} \; d_o \; L} \qquad (30)$$

R4 is the thermal resistance due to any other resistances, such as fouling and corrosion deposits.

The heat transfer coefficient on the seawater side, h_{sw}, is determined from the dimensionless group called the Nusselt number defined as follows for flow inside of tubes.

$$Nu = \frac{h_{sw}\ d_i}{k_{sw}} \qquad (31)$$

In order to find the Nusselt number, three distinct regions of flow are allowed for in the problem: laminar, transition, and turbulent flow. The division points are in terms of values of the Reynolds number, $Re = ud\rho/\mu$,

laminar: $0 < Re \leq 2{,}000$
transition: $2{,}000 < Re \leq 10{,}000$
turbulent: $10{,}000 < Re \leq$ infinity

In the laminar region the Sieder-Tate correlation is used.

$$Nu_L = 1.86\,(Re\ Pr\ d_i/L)^{1/3} \qquad (32)$$

In the turbulent region the Dittus-Boelter correlation is used.

$$Nu_{Tu} = 0.023\ Re^{0.8}\ Pr^{0.3} \qquad (33)$$

In the transition region the flow is unstable, but from data in Ref. 14 it appears that a fairly smooth curve drawn from the laminar curve to the turbulent curve adaquately represents the value of the Nusselt number in the transition region. This is done by assuming the form

$$Nu_{Tr} = y\ Nu_L + (1 - y)\ Nu_{Tu} \qquad (34)$$

where

$$y = a\ Re^3 + b\ Re^2 + c\ Re + d \qquad (35)$$

The coefficients in the polynomial are found by requiring

$$Nu_{Tr} = Nu_L \text{ at } Re = 2,000$$
$$Nu_{Tr} = Nu_{Tu} \text{ at } Re = 10,000$$

and that the first derivatives with respect to the Reynolds number match at Re equal to 2,000 and 10,000.

The I.I. Mostinski correlation, from Ref. 16, for single horizontal tube pool boiling is used to find the heat transfer coefficient on the outside of the tube. HTRI's extensive experimentation led Palen, Yarden, and Taborek, in Ref. 13, to conclude that Mostinski's equation produced the "most consistent results". His correlation is

$$h_{wf} = 0.00658\, p_c^{0.69}\, q^{0.7}\, (1.8\, r^{0.17} + 4\, r^{1.2} + 10\, r^{10}) \quad (36)$$

where

$r = p/p_c$

p_c = critical pressure of the working fluid (psia)

q = heat flux (BTU/hr-ft²)

A final heat flux relation needed is the relation for the maximum heat flux allowed. Mostinski,s correlation for a single tube is used.

$$q_{max} = 803\, p_c\, r^{0.35}\, (1 - r)^{0.9} \quad (37)$$

where the units are the same as in equation (36). From HTRI's data, under unknown conditions, q_{max} for the bundle appears to be one third of the q_{max} for a single tube.

The power required to pump the salt water through the tubes depends on the pressure drop across the tube bank. There are three causes of the pressure drop. The flow experiences a loss as the water enters into and exits from the tubes in addition to the frictional loss along the tube. From Ref. 14, the pressure drop for a liquid is determined from

$$\Delta p = 0.5 \, \rho_{sw} \, u_{sw}^{2} \left(\frac{4 f L}{d_i} + Kc - Ke \right) \tag{38}$$

In order to determine f, Ke and Kc, the flow regime must be specified. The regimes are a little different than those for the Nusselt number.

laminar: $0 < Re \le 2{,}000$
transition: $2{,}000 < Re \le 5{,}000$
turbulent: $5{,}000 < Re \le$ infinity

The Fanning friction factor, f, in the laminar regime is

$$f = 16/Re \tag{39}$$

In the turbulent regime, the Fanning friction factor is found from the Blasius equation in the range $5{,}000 < Re < 100{,}000$

$$f = 0.079/Re^{0.25} \tag{40}$$

which is a close approximation to the Karman - Nikuradse equation,

$$(4f)^{-0.5} = -0.8 + 0.87 \ln(Re(4f)^{0.5}) \qquad (41)$$

which is valid throughout the turbulent region (Re > 5,000), assuming smooth walled tubes. Either equation (40) or (41) may be used with similar results but the Blasius is better suited to this analysis because the Karman-Nikuradse equation must be solved by iteration. Equation (41) consumes more computer time to solve and can cause accuracy problems if not enough significant digits are evaluated. The friction factor in the transition region is determined by fitting a least squares curve through a hand drawn curve from the end of the laminar curve to the beginning of the turbulent curve.

The entrance effect, Kc, and the exit effect, Ke, depend on the Reynolds number, the free flow to frontal area ratio, ϕ, and, in the laminar region, on the tube length. For the boiler, ϕ is determined by

$$\phi = \frac{d_i^2 N}{Ds^2 - 4\ SA} \qquad (42)$$

Figure 9, page 7-13, in Ref. 17 is used to find Ke and Kc. The effect of Reynolds number and L/d ratio was eliminated from the relationships in the laminar and the turbulent regimes of the flow, but the Reynolds number was included in the transition region because of the large variation of Kc with Reynolds number. The following equations are used to find the entrance effect in the various flow regimes.

Laminar: (using the curve for 4 (L/d) Re = 0.1)

$$Kc_L = 1. - 0.4\ \phi \qquad (43)$$

Turbulent: (using the curve for Re = 5,000)

$$Kc_{Tu} = 0.52 - 0.4\,\phi \tag{44}$$

Transition:

$$Kc_{Tr} = \lambda\, Kc_{L} + (1 - \lambda)\, Kc_{Tu} \tag{45}$$

where

$$\lambda = e\, Re + f \tag{46}$$

The coefficients e and f are determined by requiring the transition Kc to equal the laminar Kc at Re equal to 2,000 and the turbulent Kc at Re equal to 5,000. The exit effect is the same for all flow regimes, and is found from the following equation that approximates the 4(L/d)Re = 0.05 curve in Ref. 17.

$$Ke = 1.0093 - 2.5178\,\phi + 1.1613\,\phi^{2} - 0.17677\,\phi^{3} \tag{47}$$

2. The Condenser

The design of the condenser is almost the same as the design of the boiler. It is a fixed tube sheet, tube-in-shell heat exchanger. The condenser does not need a separation volume so the entire shell is filled with tubes. In order to allow the entering vapor to reach all of the tubes, channels through the tube bank are needed. The

allowance for this is accomplished by decreasing the shell diameter by a fixed factor when calculating the number of tubes. The calculation of the working fluid side heat transfer coefficient, h , is different. The equation accounts for the condensate from one tube falling on those below it and for the subcooling of the condensate. M.M. Chen's equation, as developed in Ref. 18 and shown in a more simple form in Ref. 17, is used.

$$h_{wf} = 0.728\left[1+0.2\left(\frac{c_p \Delta T}{H_{fg}}\right)(n-1)\right]\left[\frac{g\,\rho_f\,(\rho_f-\rho_g)\,k^3\,H'_{fg}}{n\,d_o\,\mu\,\Delta T}\right]^{0.25} \tag{48}$$

where

$$h = \mathrm{BTU/ft^2\text{-}hr\text{-}^oF}$$

$$\Delta T = T_{sat} - T_w \tag{49}$$

$$H'_{fg} = H_{fg} + \frac{3}{8}\,c_p \Delta T \tag{50}$$

n = number of tubes in a column

H'_{fg} is the enthalpy corrected for subcooling of the condensate. Equation (48) replaces equation (36) in the analysis of the condenser. In the circular shaped tube bank, n is found by determining the number of tubes in a column of average height.

$$n = 9.425\,\frac{Ds}{pi} - 1 \tag{51}$$

The definition of ΔT_o becomes

$$\Delta T_o = T_{sat} - T_{sw,in} \tag{52}$$

in contrast to equation (26) above. The seawater outlet temperature is found from

$$\varepsilon = \frac{T_{sw,out} - T_{sw,in}}{\Delta T_o} \tag{53}$$

which replaces equation (25).

In order to find the heat transfer coefficient, the temperature of the outside tube surface is needed in equation (49); therefore, the thermal network is solved for the outside wall temperature and for the heat flow rate.

$$Q = UA' \; (T_w - T_{bulk}) \tag{54}$$

where

$$UA' = 1/(R1 + R2 + R4) \tag{55}$$

T_{bulk} = average of the inlet and the outlet seawater temp.

There is no factor, F, in the heat transfer equation for condensation since the effect of neighboring tubes is accounted for in equation (47).

3. The Pumps

There are three pumps in the model. Both circulation pumps for the boiler and the condenser are assumed to be propeller pumps (axial flow pumps). The feed

pumps are assumed to be centrifugal pumps. The differences in the characteristics of the two types of pump show up only in the calculation of capital costs of the pumps in the model.

The choice of the type of pump to apply is made on the basis of the flow rate, GPM, and on the head the pump is working against. Qualitatively, the propeller pump is used where the flow rates are high and the head is low. The centrifugal pumps apply where the flow rate is comparatively low and the head pressure is high. The qualitative picture is placed in better perspective through the calculation of the specific speed from

$$Nsq = \frac{RPM\ GPM^{0.5}}{Head^{0.75}} \tag{56}$$

The centrifugal pump is used when $500 < Nsq \leq 7{,}500$ and the propeller pump applies when $7{,}500 < Nsq \leq 15{,}000$, according to Ref. 19. The choice of the type of pump is checked after the optimization process has been completed by solving equation (55). If the value of the specific speed is outside of the range, $500 < Nsq < 15{,}000$, the maximum efficiency of the pump drops off rapidly; therefore, few pumps are designed to operate outside of this range.

Since the pumps are "black boxes", only the power consumed by the pumps is calculated. The power required to drive the pump is found from

$$P = \frac{M\ \Delta p}{\eta_{pump}} \tag{57}$$

D. COST FRAMEWORK

As discussed at the start of this report, the capital costs of the major components are the only costs to be considered in this model. The boiler and the condenser are both tube-in-shell heat exchangers assumed to be of the fixed tube sheet type. The only difference being that the shell of the boiler is partly filled with tubes. Both seawater circulation pumps are assumed to be propeller type pumps and made of the same material. The size and number per heat exchanger shell may be different for the boiler and the condenser. The number of working fluid feed pumps, which are the centrifugal type pumps, are assumed to equal the number of boiler shells.

The cost estimating relationships for the heat exchangers and the centrifugal pumps are taken from Ref. 20, by K.M. Guthrie. The equation for the cost of the heat exchangers is

$$\$_{he} = 118.3\ (F_d + F_p)\ F_m\ I_{he}\ (A_t)^{0.63} \qquad (58)$$

where the constants are calculated from a log-log plot of cost versus total tube surface area and

F_d = 0.8 for a fixed tube sheet
F_p = 0.0 for pressure correction ($p < 150$ psi)
F_m = material factor from Table 1
I_{he} = cost index
A_t = d_o L N

for a relevant range of $100 < A < 10{,}000$ ft².

The cost equation for the centrifugal pumps and the

associated motors is

$$\$_{fp} = 1.013\ F_m\ F_o\ I_{fp}\ C_H^{0.721} \tag{59}$$

where the constants are found from a straight line approximation of the log-log plot of cost versus C for the section of the curve from C = 30,000 to C = 300,000 for an electric motor driven pump and

F_o = 1.0 for suction temperatures < 250°F (121°C)
I_{fp} = cost index
F_m = material factor from Table 2
C_H = product of the flow rate in GPM and the pressure differential in psi

for a relevant range of 30,000 < C_H < 300,000. The cost data in Guthrie's article is given for a time base of mid-1968.

Reference 21, also by Guthrie, gives cost data for propeller pumps and centrifugal pumps. The equation for the propeller pumps with the motors is

$$\$_{cp} = 5.767\ F_m\ F_p\ I_{cp}\ \text{Flow}^{0.783} \tag{60}$$

where the constants are found from the log-log plot of cost versus flow rate for an electric motor driven pump and

F_p = 1.0 for a suction pressure < 150 psi
F_m = material factor from Table 3
I_{cp} = cost index
Flow = flow rate in GPM

for a relevant range of 1,000 < Flow < 100,000 GPM. The time base for the propeller pump data is for the end of

Table 1

MATERIAL FACTORS FOR THE HEAT EXCHANGERS

Material		
Shell	Tube	Fm
Carbon Steel	Carbon Steel	1.00
Carbon Steel	Brass	1.52
Carbon Steel	Stainless Steel	3.52
Stainless Steel	Stainless Steel	4.50
Carbon Steel	Monel	3.75
Monel	Monel	4.95
Carbon Steel	Titanium	11.10
Titanium	Titanium	16.60

Table 2

MATERIAL FACTORS FOR THE CENTRIFUGAL FEED PUMPS

Material	Fm
Cast Iron	1.00
Bronze	1.28
Cast Steel	1.32
Stainless Steel	1.93
Monel	3.23
Hastelloy C	2.89
Titanium	8.98

Table 3

MATERIAL FACTORS FOR PROPELLER PUMPS

pumps

Material	Fm
Cast Iron	1.00
Cast Steel	1.28
Stainless Steel	1.64

1970. The costs arrived at from equations (58), (59), and (60) are in dollars per component, for example, dollars per heat exchanger shell. The purpose of the cost index is allow all the equipment to be costed on a common basis. The time difference is assumed to be small enough so that the cost index is set to 1.00 for all the components.

E. ASSEMBLY OF THE MATHEMATICAL MODEL

The engineering and the cost relationships are linked together in the model. The cost equation is the function to be minimized and the implicit and explicit constraints are the engineering equations that restrict the feasible region of the problem. The explicit constraints restrict the variables, the vector X, to positive values. The rest of the model is formulated in the following form

$$\text{minimize} \quad f(X)$$

$$\text{subject to:} \quad g_i(X) \geq 0, \; i=1,2,\ldots,m$$

$$h_i(X) = 0, \; i=m+1,\ldots,m+n$$

where the definitions of the elements of X are shown in Table 4. Several of the variables must have only integer values if the system were actually built, but for the objectives of this analysis they are left as continuous real variables. For example, the number of shells may have an integer value in the actual system but this requirement can be accommodated by solving the system for a real value then integerizing the number of shells and solve the problem with the next higher and the next lower integer number of shells. The lower of the two solutions would be chosen. The final resolution of the correct number of shells would

Table 4

DEFINITION OF THE ELEMENTS OF THE VECTOR X

x(1)	=	velocity of seawater in the boiler
x(2)	=	" " " " " condenser
x(3)	=	outlet pressure in the boiler
x(4)	=	inlet " " " condenser
x(5)	=	tube length in the boiler
x(6)	=	" " " " condenser
x(7)	=	mass flow rate of the working fluid
x(8)	=	diameter of the boiler shell
x(9)	=	" " " condenser shell
x(10)	=	outside dia. of a tube in the boiler
x(11)	=	" " " " " " " condenser
x(12)	=	number of boiler shells
x(13)	=	" " condenser shells

probably depend on other than thermodynamic and cost considerations, such as a requirement that one extra shell be included to allow for one shell being down for maintenance. Obviously, the number of tubes in a shell must be an integer value but one extra tube out of several thousand is a minor matter. In addition, there may be a requirement for extra tubes to allow for a percentage to be plugged when they develop leaks so that the entire shell is not put out of commission by a minor leak in one tube.

1. Objective Function and the Constraints

The objective function is the sum of the costs of the boiler and the condenser, and the pumps. The model assumes that there is one circulating pump and one feed pump for each shell in the boiler and that there is one circulating pump for each shell in the condenser. Since there is no cost analysis for the turbine, no assumption is needed about the number of turbines in the system.

There are nineteen constraints in the present program, one of which is optional. The first three concern the saturation pressures in the boiler and the condenser. The next twelve place restrictions on various dimensions of the boiler and the condenser. The last three require that some intermediate calculations stay positive.

Using the objective function as a constraint has proved advantageous in cases where the problem was non-convex or where the program had difficulty in finding a solution. It is used by setting a boundary on the objective function so that the constraint is either infeasible or slightly feasible at the starting point. The proper choice must be found by trial and error. One caution to note is when equality constraints are involved. If the starting

point is near the optimum, the program tries to move away from the boundary and the equality constraints may not be satisfied when the program finishes.

The need for the other eighteen inequality constraints is not obvious when the problem is first formulated. The thermodynamics of the problem require, for example, that the propane saturation temperature be less than seawater temperature in order for the boiler to absorb energy, but the program has no way of knowing that, unless it it specified. If the program can improve the cost and satisfy the other constraints by reversing the relationship it will do so. This inversion may take place only for a short time during the solution of the problem, but, if the equations fail under this condition, the constraint on the temperatures in the boiler is necessary. Most of the other constraints are needed for the same reason.

Some constraints are not added to the problem since they are not binding in the solution. There are two such constraints. The first is the segmental area required to allow separation of the liquid droplets from the vapor. The segmental area set aside by the 40% rule-of-thumb is more than that required by the consideration of the physical properties of propane. The other constraint is the restriction on the maximum heat flux by Mostinski's correlation, equation (37). The constraints do not become binding in the solution.

The three equality constraints contain the calculations of the properties of the various components modeled in this analysis. The first one concerns the boiler. It states that the actual energy absorption rate of the boiler shells must be equal to the energy absorption rate required by the thermodynamic cycle. Similarly, the second equality constraint states that the heat rejected by

the condenser must equal that required by the thermodynamic cycle for the condensation of the vapor from state point 5 back to state point 1 as shown in Fig. 3. The third constraint calculates the power required by the various pumps and the power extracted from the turbine to run the system's pumps and to produce electrical power for consumption. This constraint requires that the power out of the turbine equal the power used by the pumps and the generator. The net useful power out of the system is a design objective and is specified as 25MW herein.

2. The Optimization Method

Since the arrival of the electronic computer, a great many nonlinear programming techniques have been developed from early theories and more current research. The computer program, called the Sequential Unconstrained Minimization Technique (SUMT), developed by Mylander, Holens, and McCormick in Ref. 22 is used to optimize the model. Their program implemented much of the theory contained in Ref. 23, by Fiacco and McCormick, about nonlinear programming using unconstrained minimization techniques. SUMT-Version 4 is chosen because of its ability to solve a wide variety of problems. The theoretical requirements for SUMT to be guaranteed to find a local minimum is that there exists some point which satisfies the inequality constraints and that there not be a local minimum at points where X goes to infinity. If certain convexity conditions are satisfied, then the local minimum found is a global minimum.

SUMT uses a penalty function to transform the constrained optimization problem

minimize $f(X)$

subject to: $g_i(X) \geq 0, \; i=1,2,\ldots,m$

$h_i(X) = 0, \; i=m+1,\ldots,m+n$

into a sequence of unconstrained problems of the form

$$P(X,r) = f(X) - r \sum_{i=1}^{m} \ln g_i(X) + 1/r \sum_{i=m+1}^{m+n} (h_i(X))^2 \qquad (61)$$

where $P(X,r)$ becomes the function to be minimized. The inequality constraints, $g(X)$, cause a large penalty to occur as a boundary is approached. The equality constraints add a large penalty whenever $h(X)$ departs from zero in either direction. The size of the penalties due to $g(X)$ and $h(X)$ is controlled by the arbitrary parameter "r". The algorithm solves the problem by solving a series of subproblems. For $r_1 > r_2 > r_3 \ldots > 0$, SUMT minimizes $P(X(r_n), r_n)$ in each subproblem in order to find the minimum of $f(X)$ as "r" tends to zero. When a minimum of a subproblem is found, "r" is reduced by a factor assigned by the user and the new subproblem is solved. The process continues until the convergence criteria chosen by the user is satisfied. Figure 7 illustrates schematically the results of the sequence of subproblems and the qualitative shape of the penalty function. Three methods of minimization are included in the program. The user may choose either the generalized Newton-Raphson, steepest descent, or McCormick modification of the Fletcher-Powell method.

There are several other options that control the operation of SUMT. The user may choose the initial penalty, r, or, if the problem satisfies certain conditions, the user may elect to have SUMT compute the initial "r". There are three choices for the convergence criterion of the

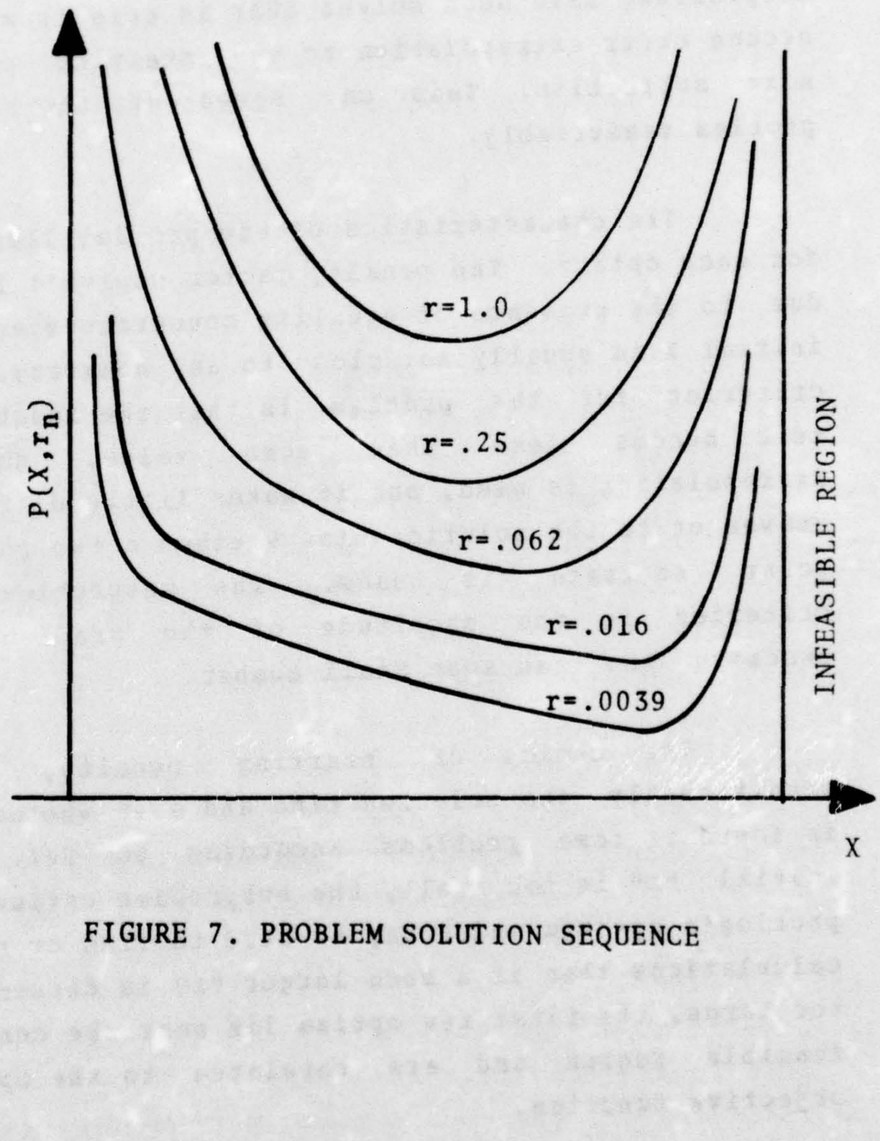


FIGURE 7. PROBLEM SOLUTION SEQUENCE

problem. The criterion used depends on the inequality penalty term being close enough to zero. Another option is the choice of the calculation of the subproblem stopping criterion . Two of the choices compute the inverse of the second partial matrix and the third choice computes only the gradient of the penalty function ($\nabla_X P(X,r)$). One of the most useful options is extrapolation. After two or three subproblems have been solved SUMT is able to make a first or second order extrapolation to the starting point for the next subproblem. This can speed up the solution of the problem measureably.

The characteristics of the problem limit the choices for each option. The penalty factor couldn't be set by SUMT due to the presence of equality constraints and because the initial X is usually not close to any boundary. The stopping criterion for the problem is that the inequality penalty term become less than some value, usually 10^{-7}. Extrapolation is used, but it makes little difference in the answer or to the solution time whether a two point or three point estimate is made. The subproblem convergence criterion is the magnitude of the gradient of $P(X,r)$ becoming less than some small number.

The choice of starting penalty, r, affects significantly the solution time and even whether a solution is found in some problems according to Ref. 7. If the initial "r" is too small, the subproblem optimum may be the problem's optimum which may be hard to find or require more calculations than if a much larger "r" is chosen. If "r" is too large, the first few optima lie near the center of the feasible region and are unrelated to the optimum of the objective function.

Scaling of the variables, constraints, and the objective function is necessary. The variables must be

within two or three orders of magnitude of each other. For example, the working fluid flow rate is of the order of 10^7 whereas the tube diameters are of the order 10^0. If the finite difference interval is 10^{-4}, the partial of the equations with respect to the mass flow rate may be lost due to the roundoff error providing the only significant figures. With the constraints and the objective function, unscaling may result in one term in the penalty function being dominant, and the algorithm finds that most of the decrease in the penalty function takes place in the dominate term. The result is that SUMT works hard to decrease that term and does not see the other terms resulting in a meaningless solution.

In making first and second order estimates of the solution, SUMT may generate negative numbers in the X vector. When it evaluates the constraints and objective function for this point, the equations fail and may cause the program to fail depending on the severity of the failure and how many times the error has occured. This problem is circumvented by testing the vector upon entry into the subroutine where the objective function and the constraints are evaluated and setting the constraint being evaluated equal to some negative constant.

The main program of SUMT is altered so that the problem can be restarted at the current solution without reading in a new problem. This is necessary because the program would give all the indications of having found a solution, but if the problem is restared it could often find a low value for the objective function. The new solution is compared to the previous solution and if the two solutions are not close enough the problem is restarted after making some changes to particular parameters, options, or constraints depending on the objective the user has in mind.

III. RESULTS AND CONCLUSIONS

One of the major tasks in the development of a nonlinear model is the linking of the model with the optimization program in order to analyze the model. The selection of the proper nonlinear programming technique is essential to the solution. Several techniques require that the objective function and the constraints be in a particular form. For example, Geometric Programming developed by Duffin, Peterson and Zener, in Ref. 24, requires that the equations be posynomials which is not true of the model developed in this paper. SUMT was chosen because of its wide applications to problems of any type. It only requires that there exist some feasible point and that local minima not exist at infinity. It has successfully solved many nonconvex problems.

Two versions of the analytical model are presented. The first is called the iterative version because the calculation of the heat absorption rate and heat rejection rate require iterating on some quantity to find them. Mostinski's correlation, equation (36), contains the heat flux which is an unknown. It is not possible to solve the system of equations explicitly for the heat flux; therefore, a successive approximation technique is used. The calculation of the heat rejected in the condenser involves the same process with the temperature of the outside tube wall as the iterating variable. SUMT is unable to arrive at a consistent solution in the time allowed, up to 90 minutes of computer time. The solutions are quite different for each try with no pattern evident about the direction the solution might lie.

There are several possible explanations for this behavior. Perhaps the behavior of this version of the problem is such that SUMT is not the best choice for the optimizing algorithm or possibly the problem cannot be solved at all in this form. Another source of problems could be the numerical differentiation procedure. Numerical differentiation is subject to severe truncation and round-off error. The subproblem always terminates when SUMT is unable to reduce the penalty function during the next iteration, from one estimate of X to the next estimate. Remember the convergence criterion, never satisfied for this version of the problem, is the magnitude of $\nabla_X P(X,r)$ becoming less than some small number. No conclusive results could be obtained with this version so the second version is used for the present analysis.

In this second version the heat transfer coefficients on the Propane side of the tube are fixed. This enables the equality constraints for the boiler and the condenser to be calculated to the full accuracy of the computer without increasing the time required. The correlations, equation (36) and (48), are used for comparison with the fixed value. If the calculated value and the fixed value are within 20 to 25% of one another, the solution would be considered to be good since this is the tolerance claimed for the correlations.

SUMT seemed to be able to solve the new problem because a series of seven solutions have been produced where the values of the elements of X and of the objective function vary no more than 3.7% and 1.2%, respectively, about the mean when different starting points and different SUMT control parameters are selected. See Table 6 for a summary of the solutions obtained from the assumed data, as shown in Table 5. The design implications of these results are quite reasonable. The range shows the spread in the solutions.

Table 5

PROBLEM INPUT DATA

Fouling Thermal Resistance Boiler and Condenser	0.0 hr-°F/BTU (0.0 °C/kW)
Elevation of Boiler Above Condenser	15 ft (4.57 m)
Channel Allowance	0.83 ft (0.25 m)
Efficiencies for the Turbine and the Pumps	85%
Seawater Temperatures - Hot	75°F (23.9°C)
- Cold	40°F (4.4°C)
Net Output Power	8.532×10^7 BTU/hr (25 MW)
Bundle Factor (F)	3.0
Cost Indexes (I)	1.00 for all comp.
Tube Wall Thickness	0.028 in (0.071 cm)
Min. Distance Between Tubes	0.5 in (1.27 cm)
Heat Transfer Coefficients for Boiler and Condenser	145 BTU/ft²-hr-°F (0.822 kW/m²-°C)

<u>Materials</u>

Boiler and Condenser	Carbon Steel Shells Brass Tubes
Feed Pumps	Bronze
Circulation Pumps	Stainless Steel

Table 6

EXAMPLE PROBLEM SOLUTION SUMMARY

Variable	Range	Mean	Var %	Units
Boiler:				
Velocity	3.323-3.347	3.335	0.7	ft/sec
	(1.013-1.020)	(1.016)		(m/sec)
Pressure	approx 0	118.2	0.0	lb/in^2
		(815.0)		(kp)
Tube length	39.16-40.00	39.71	2.1	ft
	(11.91-12.19)	(12.10)		(m)
Shell dia.	24.12-25.00	24.91	3.5	ft
	(7.352-7.620)	(7.593)		(m)
Tube dia.	0.756-0.761	0.758	0.7	in
	(1.919-1.934)	(1.926)		(cm)
# of shells	7.141-7.409	7.228	3.7	-
Condenser:				
Velocity	2.793-2.823	2.808	1.1	ft/sec
	(0.851-0.860)	(0.856)		(m/sec)
Pressure	91.53-91.60	91.56	0.1	lb/in^2
	(631.1-631.6)	(631.3)		(kp)
Tube length	39.35-40.00	39.78	1.6	ft
	(11.99-12.19)	(12.12)		(m)
Shell dia.	44.69-45.00	44.89	0.7	ft
	(13.62-13.72)	(13.68)		(m)
Tube dia.	0.7632-0.7686	0.7660	0.7	in
	(1.938-1.952)	(1.946)		(cm)
# of shells	3.297-3.401	3.331	3.1	-
Mass flow rate	22.12-22.21	22.17	0.4	10^6 lbm/hr
	(10.03-10.07)	(10.06)		(10^6 kg/hr)
System cost	7.578-7.673	7.607	1.2	10^6 $

The arithmetic mean of the seven solutions is calculated. The difference between the maximum and the minimum values is shown as a percent of the mean. The percent variation gives some idea of how well the solution was determined. Note that the tube length and shell diameter constraints are binding for both the boiler and the condenser. The subproblem still terminates only when the penalty function does not decrease from one iteration to the next. The magnitude of $\nabla_X P(X,r)$ is not becoming small enough to satisfy the convergence criterion. The value never becomes less than 0.02 and is usually greater than 0.5 when a subproblem is stopped by the program.

Useful results are obtainable, however. The engineer can learn much about his problem even from a single point. For example, from the cost breakdown in Figure 8, the portion of the costs attributable to each component can be seen. The heat exchangers, of course, dominate with 78% of the cost. The circulation pumps are about 21% of the cost with the feed pumps being 1.2% of the total cost. Since the efficiency assumed for the circulation pumps is probably too high, these pumps would be even more costly in terms of energy and money. This could indicate the priorities that should be set on the design efforts for improvements of the various components with the highest priority being the highest percentage cost component.

The system's power distribution is shown in Table 7. The significant features are the large energy absorption rate for the small useful output power and the parasitic pumping power. The lower the efficiency the greater must be the power absorbed for the same output. This is to be expected for this kind of system. The parasitic pumping power (12% of the turbine output) is about twice what it is for a fossile fuel plant.

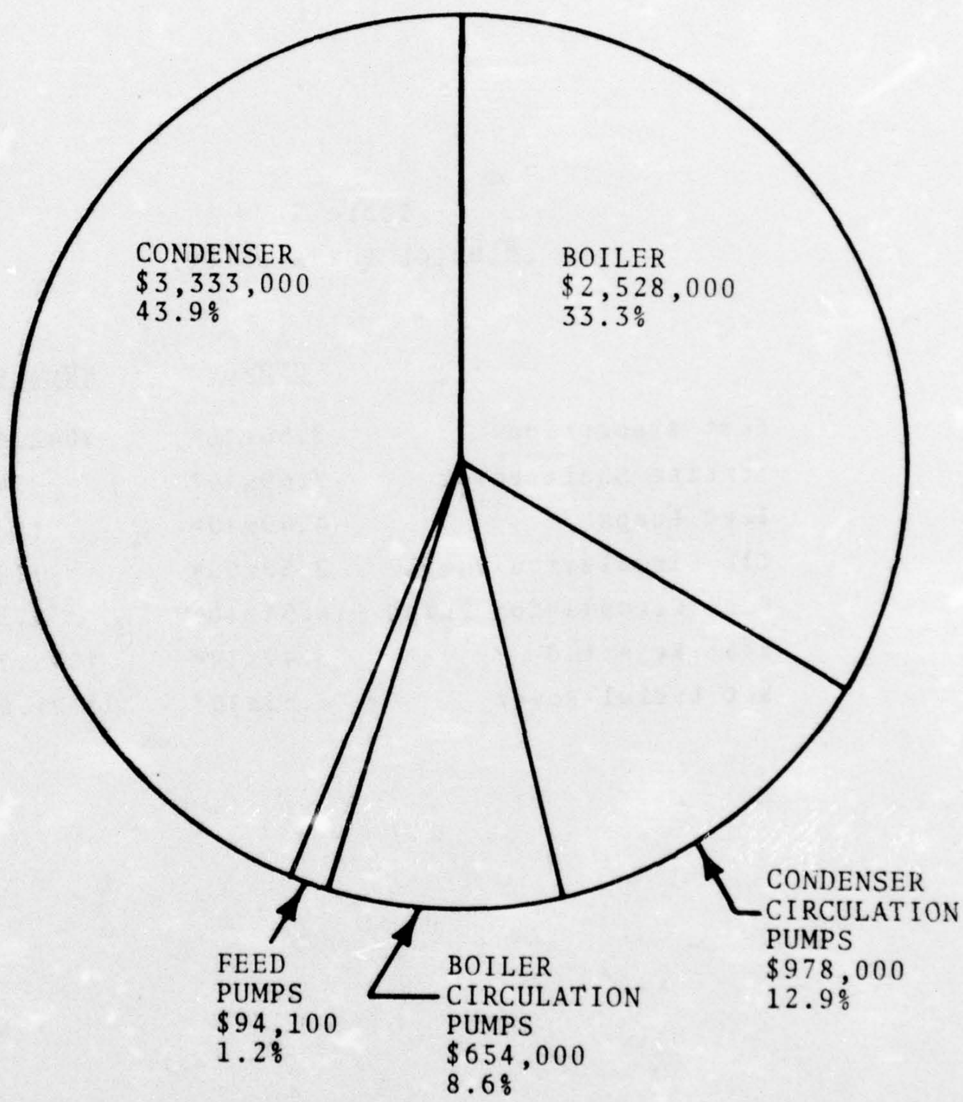


FIGURE 8. COST BREAKDOWN

Table 7

SOLUTION POWER BUDGET

	BTU/hr	MWatts
Heat Absorption	3.56×10^{9}	1042.8
Turbine Shaft Power	9.69×10^{7}	28.4
Feed Pumps	4.49×10^{6}	1.3
Elr Circulation Pumps	2.58×10^{6}	0.8
Cond Circulation Pumps	4.51×10^{6}	1.3
Heat Rejected	3.47×10^{9}	1015.7
Net Useful Power	8.53×10^{7}	25.0

Once the model has been optimized with the number of shells as real variables, the problem can be re-solved twice by fixing the shells at the next higher and the next lower integer number. For the higher number of shells, the cost increased slightly as expected, from 7.607 million dollars average to 7.983 million dollars. The problem with the lower integer number of shells could not be solved due to the constraints on the tube length and the diameter of the shells being binding before the shell numbers were integerized. The program is unable to find a new solution that satisfies the equality constraints.

Table 8 shows some miscellaneous data from the model that deserves comment. The overall heat transfer coefficient of the boiler is approximately three times that for the condenser which is the major reason for the lower cost of the boiler relative to the condenser. However, remember that the bundle factor, equal to 3, is a part of the heat transfer coefficient. More data must be gotten about this factor in order to improve the reliance on any solution. There is a significant difference between the Propane heat transfer coefficients that were assumed and the theoretical values calculated from the correlations. Other values should be tried until the fixed value is within the 20% tolerance on the theoretical value. The flow of seawater through the tubes in both the boiler and the condenser is turbulent. Apparently, the resulting cost savings due to the improved heat transfer rate through a smaller area is more than the increased costs of the higher pumping power needed. It seems likely that the solution obtained is globally optimal; several different starting points were tried. The problem has been started in both the laminar and the turbulent flow regions with no change in the results.

Table 8
ADDITIONAL DATA FROM SAMPLE
PROBLEM SOLUTION

	Boiler	Condenser	Units
Working Fluid Heat Transfer Coefficients			
Assumed	145.	145.	BTU/hr-ft²-°F
	(0.822)	(0.822)	(kW/m²-°C)
Theoretical	227.	241.	
	(1.29)	(1.37)	
Surface Area per Shell	2.39×10^5	1.27×10^6	ft²
	(2.22×10^4)	(1.18×10^5)	(m²)
Seawater Outlet Temp	72.3	43.8	°F
	(22.4)	(6.55)	(°C)
Shell Saturation Temp	66.7	49.9	°F
	(19.3)	(9.94)	(°C)
Seawater Mass Flow Rate	4.39×10^8	9.11×10^9	lbm/hr
	(1.99×10^8)	(4.13×10^9)	(kg/hr)
Mean Temp Difference	6.67	7.85	°F
	(3.71)	(4.36)	(°C)
Overall Heat Transfer	310.6	105.0	BTU/hr-ft²-°F
Coefficient	(1.76)	(0.595)	(kW/m²-°C)
Heat Flux	2070.0	824.9	BTU/ft²-hr
	(6.528)	(2.602)	(kW/m²)
Seawater Reynolds no.	19,800.	9,960.	-

IV. RECOMMENDATIONS

The present model has a great deal more information to offer, but some improvements in the model should be made to improve the ability of a nonlinear program to solve the problem. The first recommendation would be to study the equality constraints and place inequality constraints on any variable or grouping of variables if they both can become negative and also must have their logarithms computed. Examples, already included in the program, are the number of tubes in the shells. The heat transfer coefficients for the working fluid side in the boiler and the condenser could be included as equality constraints. This can be done by adding the heat transfer coefficients to the vector X and requiring that the variables be equal to the coefficients as computed by the appropriate correlation. The same could be done with Palen and Small's property-dependent calculation of the separation area by equation (12) and (13). From an overall standpoint, the results seem to indicate either the problem is too difficult to be solved efficiently in its present form or that another nonlinear program, other than SUMT, might be better suited to the problem. Numerical differentiation of the constraints in this problem do not seem to help the program to find the solution since the magnitude of $\nabla_X P(X,r)$ is not near zero when a subproblem stops.

Once the optimization scheme is functioning properly, the model can be used to study the effects of fouling and corrosion and the costs of their prevention. Many proposals have been made that must be analyzed in the context of the total system. Some such as Amertap, where plastic foam balls are pumped through the tubes to clean the heat

transfer surfaces, add a cost penalty from the capital and operating cost of the Amertap system and from the increased pumping power required. Coatings add the capital costs of the coating and the added capital cost of increased heat transfer area required because of the increased thermal resistance of the coating.

A sensitivity analysis can be performed on the model to test the assumptions and the various constants. The sensitivity to changes in these factors can indicate how much one should be willing to pay for the improvement sought. High sensitivity indicates areas where the greatest effort for improvement should be desired or the priorities should be set on research into the unknown areas of the problem. The trade-off between cost and performance can be studied by changing the output power and looking for the lowest cost per kilowatt-hour, which is the measure by which conventional power plants are compared. One could look into the cost and the benefits of increasing the depth from which the cold water is pumped versus the improved temperature differential possible from using the colder water.

For the capital cost of the system to be complete, the turbine design must be included and cost estimates made. The turbine will be unique due to the very low pressure potential drop it must use. Some work has been done by other groups working on OTEC, Ref. 25 and 26. Reference 27, by Robert L. Bartlett, can provide costing and design information although he deals only with steam turbines. The remaining part of the system that is desireable for inclusion in the analysis presented herein is the cold seawater intake pipe. It will be on the order of 1,000 ft (304.8m) long and 100 ft (30.4 m) in diameter.

APPENDIX A

```
C**********        MAIN        *****************
C              MARCH 1971
C
C MAIN IS THE PROGRAM THAT INITIATES THE SUMT ALGORITHM. THE INPUT OF
C PARAMETERS, OPTIONS, AND STARTING POINT IS DONE IN MAIN. AFTER THE
C SOLUTION OF ONE NLP PROBLEM MAIN LOOKS FOR DATA FOR ANOTHER NLP PROB.
      IMPLICIT REAL*8(A-H,O-Z,$)
      COMMON /SHARE/ X(100),DEL(100),A(100,100),N,M,MN,NP1,NM1
      COMMON /EQAL/ H,H1,MZ
      COMMON /OPTNS/ NT1,NT2,NT3,NT4,NT5,NT6,NT7,NT8,NT9,NT10
      COMMON /VALUE/ F,G,PO,RSIGMA,RJ(200),RHO
      COMMON /CRST/ DELX(100),DELXO(100),RHOIN,RATIO,EPSI,THETAO,RSIG1,G
     1,X1(100),X2(100),X3(100),XR2(100),XR1(100),PR1,PR2,F1,F1,RJ1(200)
     2,CCTT,PGRAD(100),DIAG(100),PREV2,ADELX,NTCTR,NUMINI,NPHASE,NSATIS
      COMMON /EXPOPT/ NEXCP1,NEXCP2,XEP1,XEP2
      COMMON /DIMEN/ VL,SA,HSC,HMAX,PITCH,OD,DI,CLEAR,DS,S,TUBEL,HS,HEIG
     1HT,AAVG,ATOT,ATOTC,BANKHT,HTMAX,ATOTB,THICK,CN,COD,CID,CTUBL,CPITC
     2H,CDS,CCLR,CTHK,CS,NUF,NCN,NMAX,INDEX,NI,NIC
      COMMON /COST/ $BLR,$FDPMP,$BCIRC,$CCND,$CCIFC,$TCT
      COMMON /SHELL/ AREAFR
      COMMON /MISC1/ N1,N2,N3
      COMMON /$/ $X
      COMMON /AECDE/ ASTOP
C     PARAMETER CARD
      MNI = 1
      MNN = 1
      $Y = 0.C
   20 READ (5,80,END=60) EPSI,RHOIN,THETAO,RATIO,TMMAX,M,N,MZ
      READ (5,90) (X(I),I=1,N)
      CALL READIN
      READ (5,110) NT1,NT2,NT3,NT4,NT5,NT6,NT7,NT8,NT9,NT10
      READ (5,90) XEP1,XEP2
      READ (5,110) NEXCP1,NEXCP2
   30 CONTINUE
      N1 = M+1
      N2 = M+2
      N3 = M+3
      CALL SET (TMMAX)
C     INITIAL  X VERCTOR CARD   FORMAT
```

```
      NTCTR = 0
      NP1 = N+1
      NM1 = N-1
C SUBROUTINE READIN IS UNDER PROGRAMMER CONTROL
C OPTION CARD FOLLOWS PROGRAMMERS DATA
      WRITE (6,140)
      WRITE (6,150) N,M,MZ,TMMAX,RHCIN,RATIO,EPSI,THETAO
      WRITE (6,160)
      WRITE (6,110) NT1,NT2,NT3,NT4,NT5,NT6,NT7,NT8,NT9,NT10
C--READ TOLERANCES
      WRITE (6,120)
      WRITE (6,100) XEP1,XEP2
C--READ SECOND OPTION CARD
      WRITE (6,130)
      WRITE (6,110) NEXOP1,NEXOP2
      CALL TIMEC
      NPHASE = 4
      AFR = 6.*AREAFR
      WRITE (6,70) AFR
C --- JUST TO GET AN INITIAL PRINTOUT
      CALL OUTA
      $X = $TCT/1.E07+.0001
      CALL EVALU
      FC = 0.0
      G = 0.0
      F = 0.0
      RSIGMA = 0.0
      CALL OUTPUT (2)
      CALL STORE
      IF (NEXOP1.GT.1) CALL CHCKER
      IF (NEXOP1.EQ.3) STOP 01072
      IF (NEXOP1.EQ.5) STOP 01104
      CALL FEAS
C NPHASE=5 IS USED TO INDICATE NO FEASIBLE POINT EXIST
      GO TO (40,40,40,40,20), NPHASE
   40 NPHASE = 2
      NTCTR = 0
      CALL BODY
C
C   RESTART THE CURRENT PROBLEM IF THE LAST SOLUTION IS AN IMPROVEMENT
C   OVER THE PREVIOUS SOLUTION. ALSO MOVE THE BOUNDARY OF THE OBJ FUNC
C   CONSTRAINT CLOSE TO THE PRESENT  SOLUTION.
C
      IF (DABS($TCT-$Y).LE.$TCT/1.E05) GO TO 50
      $Y = $TCT
      $X = $TCT/1.E07+.0001
      MNN = MNN+1
      GO TO 30
```

```
 90 CONTINUE                                                              MAIN 880
    CALL OUTA                                                             MAIN 890
    GO TO 20                                                              MAIN 900
 60 STOP                                                                  MAIN 910
C                                                                         MAIN 920
C   PARAMETER CARD                                                        MAIN 930
C   INITIAL  X VERCTOR CARD   FORMAT                                      MAIN 940
C   OPTION CARD FORMAT                                                    MAIN 950
C   OPTION CARD FORMAT                                                    MAIN 960
C                                                                         MAIN 970
C                                                                         MAIN 980
C                                                                         MAIN 990
 70 FORMAT ('OTUBE BUNDLE WALL CLEARANCE=',F8.4,' (IN)')                  MAIN1000
 80 FORMAT (5E12.0,3I4)                                                   MAIN1010
 90 FORMAT (6E12.6)                                                       MAIN1020
100 FORMAT (6E20.7)                                                       MAIN1030
110 FORMAT (10I7)                                                         MAIN1040
120 FORMAT (13HO TOLERANCES )                                             MAIN1050
130 FORMAT (26HO SECOND SET OF OPTIONS       )                            MAIN1060
140 FORMAT (56H1 NONLINEAR PROGRAMMING ROUTINE-SUMT VERSION 4   3/22/71MAIN1070
   1 )                                                                    MAIN1080
150 FORMAT (1HO,5X,2HN=I3,6X,2HM=I3,6X,3HMZ=I3//8X,10HMAX. TIME=E14.7,MAIN1090
   14X,2HR=E14.7,4X,6HRATIO=E14.7,6X,8HEPSILON=E14.7,4X,6HTHETA=E14.7)MAIN1100
160 FORMAT (18HO OPTIONS SELECTED)                                        MAIN1110
    END                                                                   MAIN1120
```

```
      SUBROUTINE RESTNT (J,VAL)
C
C  THIS SUBROUTINE EVALUATES THE OBJECTIVE FUNCTION AND THE
C  CONSTRAINTS ONE AT A TIME AS REQUESTED BY THE SUMT PROGRAM.
C
C
      IMPLICIT REAL*8(A-H,O-Z,$)
      COMMON /SHARE/ X(100),DEL(100),AX(100,100),N,M,MN,NP1,NM1
      COMMON /DIMEN/ VL,SA,HSC,HMAX,PITCH,OD,DI,CLEAR,DS,S,TUBEL,HS,HEIG
     1HT,HAVG,ATOT,ATOTC,BANKHT,HTMAX,ATOTB,THICK,CN,COD,CID,CTUBL,CPITC
     2H,CDS,CCLR,CTHK,CS,NUP,NDN,NMAX,INDEX,NI,NIC
      COMMON /FLOW/ FLOWF,FLOSW,USW,RESW,UINF,UMAX,REAVG,FLOSWT,GPMWF,GP
     1MSW,CFLOSW,CUSW,CRESW,CGPMSW,CFLSWT
      COMMON /RESIST/ R1,R2,R4,R3C,R3E,HC,HCSW,HB,FACT,CWF,CSW,CMAX,CMIN
     1,LAC,NTLC,UAB,NTUB,RAT,GAMMA,EPS,CR1,CR2,CR3,CR4,CHFG,CHCSW,CLA,CN
     2TL,CCSW,CHSW
      COMMON /PROP/ RHOSWH,RHOSWC,VISSWH,VISSWC,CKSWH,CKSWC,PRSWH,PRSWC,
     1CPSWH,CPSWC,CKMET,BKMET,PCRIT,FMBLR,FMCIRC,FMFD,FMCOND,DEXBLR,DEXC
     2RC,DEXFD,DEXCND
      COMMON /PRESS/ PO,P,R,BKE,KC,RHC,F,DELTAP,HEAD,HEADWF,DELPWF,CKC,C
     1F,CDELF,CHD,CRHO,CKE,FCOND
      COMMON /TEMPER/ DELTOC,BTSWIN,TSWOC,DELTOE,TSWOB,DELT,TWALL,CTSWIN
     1,CTWOUT,CDELT
      COMMON /HEAT/ QTUBEC,QTUBEB,CSTUBB,CTEMP,CTTUBE,QTOT,CQTOT,QSTUBC,
     1CREJ,NETPWR,PWPWF,PWPSW,EFFPWF,EFFTUR,EFFFSW,PWRTUR,CMAX,SYSEFF,CC
     2REJ,CPWFSW,CS,CQTUBE,NET,CYCEFF
      COMMON /CYCLE/ H1,H2,H4,H5,H5S,T1,T2
      COMMON /MISC/ MSG1,MSG2,MSG3,MSG4,MSG5,MSG6,MSG7
      COMMON /COST/ $BLR,$FDPMP,$BCIRC,$CCND,$CCIRC,$TOT
      COMMON /SIM/ A(16),B(4)
      COMMON /MISCEL/ HT
      COMMON /TUBEN/ BNI,CNI
      COMMON /SHELL/ AREAFR
      COMMON /$/ $X
      COMMON /SSS/ HBA,CHCA
      REAL *8MSG1,MSG2,MSG3,MSG4(2),MSG5(2),MSG6(2),MSG7(2),STRG
      REAL *8KC,NTUC,NTUB,NU,NETPWR,NET,NETOUT
C
C
C  SET THE VARIABLES
C
      CALL SETV
C
C IF ANY VARIABLE IS NEGATIVE, BYPASS THE CONSTRAINTS.
C
      IF (USW.LE.0.0.OR.CUSW.LE.0.0.OR.TUBEL.LE.0.0.OR.CTUBL.LE.0.0.OR.F
     1LOWF.LE.0.0.OR.DS.LE.0.0.OR.CDS.LE.0.0.OR.OD.LE.0.0.OR.COD.LE.0.0.
     2OR.S.LE.0.0.OR.CS.LE.0.0) GO TO 230
```

```
C
      DI = OD-2.0*THICK
      CID = COD-2.0*CTHK
C
C
C     GO TO THE REQUESTED OBJ FUNC OR CONSTRAINT.
C
      IF (J.EQ.0) GO TO 30
      IF (M.EQ.15) GO TO 20
      GO TO (200,210,220,240,250,260,270,280,290,300,310,320,330,340,350
     1,360,370,380,50,90,130), J
   20 GO TO (200,210,220,240,250,260,270,280,290,300,310,320,330,340,350
     1,360,370,380,30,50,90,130), J
C.......................................................................
C
C
C          OBJECTIVE FUNCTION
C
C
C.......................................................................
   30 RHOFPC = RHOF(PCOND)
      DELPWF = PO+6.94444E-03*RHOFPC*HT-PCOND
      HEADWF = 144.*DELPWF/RHOFPC
      CALL AREA (1,MM)
      CALL AREA (2,MM)
C
C     COMPUTE THE VARIABLES FOR THE COST FUNCTIONS
C
      FLOSW = 19.63495*RHOSWH*USW*DI**2
      CFLOSW = 19.63495*RHOSWC*CUSW*CID**2
      ATOTB = .2618*OD*TUBEL*ENI
      CGPMSW = CFLOSW*.124722*CNI/RHOSWC
      ATOTC = .2618*COD*CTUBL*CNI
      GPMWF = FLOWF*.124722/(RHOFPC*S)
      GPMSW = FLOSW*.124722*BNI/RHOSWH
      CH = GPMWF*DELPWF
C
C     THE COST FUNCTIONS
C
      $BLR = 94.64*FMBLR*DEXBLR*ATOTB**.63
      $BCIRC = 5.767*FMCIRC*DEXCRC*GPMSW**.783
      $FDPMP = 1.013*FMFD*DEXFD*CH**.721
      $CCIRC = 5.767*FMCIRC*DEXCRC*CGPMSW**.783
      $CCND = 94.64*FMCOND*DEXCND*ATOTC**.63
      VAL = ($BLR*S+$FDPMP*S+$BCIRC*S+$CCND*CS+$CCIRC*CS)/1.E07
C
C     IF OBJ FUNC IS A CONSTRAINT EVALUATE THE CONSTRAINTS VALUE
C
```

```
      IF (J.EQ.19) GO TO 40
      GO TO 390
   40 VAL = SX-VAL
      GO TO 390
C.......................................................................
C
C
C        CONSTRAINTS
C
C
C.......................................................................
C
C
C        EQUALITY CONSTRAINT 1: BOILER
C
C
C.......................................................................
   50 RESW = 300.*DI*USW*RHOSWH/VISSWH
      RHOFPC = RHOF(PCOND)
      RHOFP = RHOF(PO)
      X2 = PRSWH**.3
      HFPC = HF(PCOND)
      H1 = HFPC
      H4 = HG(PO)
C
C  FIND THE ENTHALPY OF STATE POINT 2
C
      DELPWF = PO+6.94444E-03*RHOFPC*HT-PCOND
      HEADWF = 144.*DELPWF/RHOFPC
      H2 = H1+HEADWF/(778.16*EFFPWF)
C
C   FIND THE NUMBER OF TUBES IN THE BOILER
C
      CALL AREA (1,MM)
C
C   FIND THE HEAT TRANSFER COEFFICIENT FOR THE SEAWATER
C
      X1 = (PRSWH*DI/(TUBEL*12.))**.3333333
      IF (RESW.LE.2000..OR.RESW.GE.10000.) GO TO 60
      CALL SETA (DI,TUBEL,PRSWH)
      CALL CSIMQ (A,B,4,KS)
      PCT = B(4)+RESW*(B(3)+RESW*(B(2)+RESW*B(1)))
      NU = 2.02*PCT*X1*RESW**.333333+(1.-PCT)*.023*X2*RESW**.8
      IF (NU.LE.0.0) WRITE (6,400) NU
      GO TO 80
   60 IF (RESW.LE.2000.) GO TO 70
      NU = .023*RESW**.8*X2
      GO TO 80
```

```
   70 NU = 1.86*RESW**.333333*X1                                         REST1450
      IF (NU.LE.3.66) NU = 3.66                                          REST1460
   80 HCSW = 12.*NU*CKSWH/DI                                             REST1470
C                                                                        REST1480
C     COMPUTE THE VARIOUS THERMAL RESISTANCES                            REST1490
C                                                                        REST1500
      R1 = 3.81972/(HCSW*DI*TUBEL)                                       REST1510
      R2 = .159155*DLOG(OD/DI)/(BKMET*TUBEL)                             REST1520
      FLOSW = 19.63495*RHOSWH*USW*DI**2                                  REST1530
      CSW = FLOSW*CPSWH                                                  REST1540
      HEIGHT = HS+.5*DS                                                  REST1550
      P = PO+3.47222E-03*RHOFP*HEIGHT                                    REST1560
      R = P/PCRIT                                                        REST1570
      FLOSWT = FLOSW*BNI*S                                               REST1580
      AA = OD*TUBEL                                                      REST1590
      AB = .0168131*PCRIT**.69                                           REST1600
      AC = 1.8*R**.17+4.*R**1.2+10.*R**10                                REST1610
      AD = R1+R2+R4                                                      REST1620
      AE = AB*AC/AA**.7                                                  REST1630
      FACT = 3.0                                                         REST1640
      DELTCB = BTSWIN-TSAT(P)                                            REST1650
      HBA = 145.                                                         REST1660
      R3B = 3.81972/(HBA*AA)                                             REST1670
C                                                                        REST1680
C      FIND THE TOTAL HEAT TRANSFER RATE AND THE SEAWATER OUTLET TEMP    REST1690
C                                                                        REST1700
      UAB = 1./(AD+R3B)                                                  REST1710
      NTUB = UAB/CSW                                                     REST1720
      TSWOB = BTSWIN-DELTCB*(1.0-DEXP(-NTUB))                            REST1730
      QTUBEB = UAB*FACT*DELTCB*(1.0-DEXP(-NTUB))/NTUB                    REST1740
      HB = AE*QTUBEB**.7                                                 REST1750
      DELT = (BTSWIN+TSWOB)/2.-T2                                        REST1760
      QQTOT = QTUBEB*S*BNI                                               REST1770
C                                                                        REST1780
C     FIND THE  THERMODYNAMIC CYCLE HEAT ABSORPTION RATE                 REST1790
C                                                                        REST1800
      QTOT = FLOWF*(H4-H2)                                               REST1810
C                                                                        REST1820
C     CALCULATE THE CONSTRAINT                                           REST1830
C                                                                        REST1840
      VAL = (QQTOT-QTOT)/1.E09                                           REST1850
      GO TO 390                                                          REST1860
```

```
C.......................................................................
C
C
C          EQUALITY CONSTRAINT 2: CONDENSER
C
C
C.......................................................................
   90 CRESW = 300.*CID*CUSW*RHOSWC/VISSWC
      T1 = TSAT(PCOND)
      T2 = TSAT(PO)
      RHOGPC = RHOG(PCOND)
      RHOFPC = RHOF(PCOND)
      HFPC = HF(PCOND)
      H4 = HG(PC)
      HGPC = HG(PCOND)
      H1 = HFPC
C
C     FIND THE ENTHALPY OF STATE POINT 5
C
      H5S = H1+(ENTG(T2)-ENTF(T1))*(HGPC-HFPC)/(ENTG(T1)-ENTF(T1))
      H5 = H4-EFFTUR*(H4-H5S)
C
C     FIND THE NUMBER OF TUBES IN THE CONDENSER
C
      CALL AREA (2,MM)
C
C     FIND THE SEAWATER HEAT TRANSFER COEFFICIENT
C
      CX1 = (PRSWC*CID/(CTUBL*12.))**.333333
      CX2 = PRSWC**.3
      IF (CRESW.LE.2000..OR.CRESW.GE.10000.) GO TO 100
      CALL SETA (CID,CTUBL,PRSWC)
      CALL DSIMQ (A,B,4,KT)
      CPCT = B(4)+CRESW*(B(3)+CRESW*(B(2)+CRESW*B(1)))
      CNU = 2.02*CPCT*CX1*CRESW**.333333+(1.-CPCT)*.023*CX2*CRESW**.8
      IF (CNU.LE.0.0) WRITE (6,410) CNU
      GO TO 120
  100 IF (CRESW.LE.2000.) GO TO 110
      CNU = .023*CRESW**.8*CX2
      GO TO 120
  110 CNU = 2.02*CRESW**.333333*CX1
      IF (CNU.LE.3.66) CNU=3.66
  120 CHSW = 12.*CNU*CKSWC/CID
C
C     FIND THE THERMAL RESISTANCES
C
      CR1 = 3.81972/(CHSW*CID*CTUBL)
      CR2 = .159155*DLOG(COD/CID)/(CKMET*CTUBL)
```

```
      CFLOSW = 19.63495*RHOSWC*CUSW*CID**2
      CFLSWT = CFLOSW*CNI*CS
      CCSW = CFLOSW*CPSWC
      CN = 9.42478*CDS/CPITCH-1.0
      HFG = HGPC-HFPC
      BB = CPF(T1)
      BC = CR1+CR2
      BD = BC+CR4
      BF = .2*BB*(CN-1.)/HFG
      BG = RHOFPC*1391040.*(RHOFPC-RHOGPC)*TKF(T1)**3
      BH = CN*COD*VISCF(T1)
      BE = BG/BH
      BI = T1-CTSWIN
      BJ = COD*CTUBL
      BK = 3.*BB/8.
      CHCA = 145.
      CR3 = 3.81972/(CHCA*BJ)
C
C  FIND THE HEAT TRANSFER RATE, THE SEAWATER OUTLET TEMP,AND THE
C  OUTTER TUBE WALL TEMP
C
      CUA = 1./(BD+CR3)
      CNTU = CUA/CCSW
      D = BI*(1.0-DEXP(-CNTU))
      CQTUBE = CCSW*D
      CTWOUT = CTSWIN+D
      TWALL = (CTSWIN+CTWOUT)/2.+CQTUBE*BC
      CDELT = T1-TWALL
      CHFG = HFG+BK*CDELT
      C = 1.+BF*CDELT
      CHOSW = .728*C*(BE*CHFG/CDELT)**.25
      CQREJ = CQTUBE*CS*CNI
C
C  FIND THE THERMODYNAMIC HEAT TRANSFER RATE
C
      QREJ = FLOWF*(H5-H1)
C
C  COMPUTE THE CONSTRAINT
C
      VAL = (CQREJ-QREJ)/1.E09
      GO TO 390
```

```
C......................................................................
C
C
C       EQUALITY CONSTRAINT 3: NET POWER CUT
C
C
C......................................................................
  130 RESW = 300.*DI*USW*RHOSWH/VISSWH
      CRESW = 300.*CID*CUSW*RHOSWC/VISSWC
      T1 = TSAT(PCOND)
      T2 = TSAT(PC)
      RHOFPC = RHOF(PCOND)
      HFPC = HF(PCOND)
      HGPC = HG(PCOND)
      H4 = HG(PC)
      H1 = HFPC
C
C FIND THE ENTHALPY AT STATE POINTS 5S AND 2
C
      DELPWF = PC+6.94444E-03*RHOFPC*HT-PCOND
      HEADWF = 144.*DELPWF/RHOFPC
      H5S = H1+(ENTG(T2)-ENTF(T1))*(HGPC-HFPC)/(ENTG(T1)-ENTF(T1))
      H2 = H1+HEADWF/(778.16*EFFPWF)
C
C  FIND THE NUMBER OF TUBES IN THE BOILER AND THE CONDENSER
C
      CALL AREA (1,MM)
      CALL AREA (2,MM)
C
C  DETERMINE THE POWER REQUIRED FOR THE FEED PUMPS
C
      X2 = PRSWH**.3
      X1 = (PRSWH*DI/(TUBEL*12.))**.3333333
      PWPWF = FLOWF*(H2-H1)
C
C  FIND THE POWER REQUIRED FOR THE BOILER CIRC PUMPS
C
      RHO = DI**2*BNI/(144.*(DS**2-1.27324*SA))
      IF (RHO.LE.0.0) WRITE (6,420) RHO
      BKE = 1.00929+RHO*(-2.51782+RHO*(1.16126-.176768*RHO))
      IF (RESW.GT.2000.) GO TO 140
      F = 16./RESW
      KC = 1.-.4*RHO
      GO TO 160
  140 IF (RESW.LT.5000.) GO TO 150
      F = .079/RESW**.25
      KC = .52-.4*RHO
      GO TO 160
```

```
150 LAMBDA = 1.666667-3.333333E-04*RESW
    KC = .52+.48*LAMBDA-.4*RHO
    F = 1.2948E-01-1.71438E-04*RESW+9.05226E-08*RESW**2-2.2345E-11*RES
   1W**3+2.5969E-15*RESW**4-1.1393E-19*RESW**5
160 DELTAP = 1.07833E-04*RHCSWH*(USW**2)*(48.*F*TUBEL/DI+KC-BKE)
    PWPSW = 4.6263*USW*DI**2*S*DELTAP*BNI/EFFPSW
C
C  FIND THE POWER REQUIRED FOR THE CONDENSER CIRC PUMPS
C
    CRHC = CID**2*CNI/(144.*(CDS**2))
    CKE = 1.00929+CRHO*(-2.51782+CRHC*(1.16126-.176768*CRHO))
    IF (CRESW.GT.2000.) GO TO 170
    CF = 16./CRESW
    CKC = 1.-.4*CRHO
    GO TO 190
170 IF (CRESW.LT.5000.) GO TO 180
    CF = .079/CRESW**.25
    CKC = .52-.4*CRHO
    GO TO 190
180 CLMDA = 1.666667-3.333333E-04*CRESW
    CKC = .52+.48*CLMDA-.4*CRHO
    CF = 1.2948E-01-1.71438E-04*CRESW+9.05226E-08*CRESW**2-2.2345E-11*
   1CRESW**3+2.5969E-15*CRESW**4-1.1393E-19*CRESW**5
190 CDELP = 1.07833E-04*RHCSWC*(CUSW**2)*(48.*CF*CTUBL/CID+CKC-CKE)
    CPWPSW = 4.6263*CUSW*CID**2*CDELP*CNI*CS/EFFPSW
C
C  FIND THE POWER PROVIDED BY THE TURBINE
C
    PWRTUR = EFFTUR*FLOWF*(H4-H5S)
C
C  EVALUATE THE CONSTRAINT
C
    VAL = (PWRTUR-NETPWR-PWPSW-PWPWF-CPWPSW)/1.E07
    GO TO 390
C.......................................................................
C
C
C        INEQUALITY CONSTRAINTS
C
C
C.......................................................................
200 T2 = TSAT(PO)
    VAL = BTSWIN-T2
    GO TO 390
210 T1 = TSAT(PCOND)
    VAL = T1-CTSWIN
    GO TO 390
220 T1 = TSAT(PCOND)
```

```
      T2 = TSAT(PO)
      VAL = T2-T1
      GO TO 390
  230 VAL = -1.0
      WRITE (6,430) (X(K),K=1,N)
      GO TO 390
  240 VAL = 40.-TUBEL
      GO TO 390
  250 VAL = 40.-CTUBL
      GO TO 390
  260 VAL = CID-.25
      GO TO 390
  270 VAL = DI-.25
      GO TO 390
  280 VAL = (S-2.)/10.
      GO TO 390
  290 VAL = (CS-2.)/10.
      GO TO 390
  300 VAL = TUBEL-3.
      GO TO 390
  310 VAL = CTUBL-3.
      GO TO 390
  320 VAL = 25.-DS
      GO TO 390
  330 VAL = 45.-CDS
      GO TO 390
  340 VAL = 155.-S
      GO TO 390
  350 VAL = 155.-CS
      GO TO 390
  360 CALL AREA (1,MM)
      VAL = BNI/1000.
      GO TO 390
  370 RHOFP = RHOF(PO)
      HEIGHT = .6*DS
      P = PO+3.47222E-03*RHOFP*HEIGHT
      DELTOB = BTSWIN-TSAT(P)
      VAL = DELTOB
      GO TO 390
  380 CALL AREA (2,MM)
      VAL = CNI/1000.
      GO TO 390
  390 CONTINUE
      RETURN
C
C
C
  400 FORMAT (' NU=',E16.4)
```

```
  410 FORMAT (' CNU=',E16.4)                                          REST4200
  420 FORMAT (' RHO=',E16.4)                                          REST4210
  430 FORMAT ('ONON-NEGATIVITY CONSTRAINT VIOLATED:',(7F9.4))          REST4220
      END                                                             REST4230
```

```
      SUBROUTINE READIN
C
C   THIS SUBROUTINE READS IN THE NECESSARY CONSTANTS AND FINDS THE
C   PROPERTIES OF SEAWATER.
C
      IMPLICIT REAL*8(A-H,O-Z,$)
      COMMON /SHARE/ X(100),DEL(100),A(100,100),N,M,MN,NP1,NM1
      COMMON /DIMEN/ VL,SA,HSC,HMAX,PITCH,OD,DI,CLEAR,DS,S,TUBEL,HS,HEIG
     1HT,AAVG,ATOT,ATOTC,BANKHT,HTMAX,ATCTB,THICK,CN,COD,CID,CTUBL,CPITC
     2H,CDS,CCLR,CTHK,CS,NUP,NDN,NMAX,INDEX,NI,NIC
      COMMON /FLOW/ FLOWF,FLOSW,USW,RESW,UINF,UMAX,REAVG,FLOSWT,GPMWF,GP
     1MSW,CFLOSW,CUSW,CRESW,CGPMSW,CFLSWT
      COMMON /RESIST/ R1,R2,R4,R3C,R3B,HC,HCSW,HB,FACT,CWF,CSW,CMAX,CMIN
     1,LAC,NTUC,UAB,NTUB,RAT,GAMMA,EPS,CR1,CR2,CR3,CR4,CHFG,CHCSW,CLA,CN
     2TU,CCSW,CHSW
      COMMON /PROP/ RHOSWH,RHOSWC,VISSWH,VISSWC,CKSWH,CKSWC,PRSWH,PRSWC,
     1CPSWH,CPSWC,CKMET,BKMET,PCRIT,FMBLR,FMCIRC,FMFD,FMCOND,DEXBLR,DEXC
     2RC,DEXFD,DEXCND
      COMMON /PRESS/ PO,P,R,BKE,KC,RHC,F,DELTAP,HEAD,HEADWF,DELPWF,CKC,C
     1F,CDELF,CHD,CRHO,CKE,PCOND
      COMMON /TEMPER/ DELTOC,BTSWIN,TSWOC,DELTOB,TSWOB,DELT,TWALL,CTSWIN
     1,CTWOUT,CDELT
      COMMON /HEAT/ QTUBEC,QTUBEB,CSTUBE,CTEMP,CTTUBE,CTOT,CCTOT,CSTUBC,
     1QREJ,NETPWR,PWPWF,PWPSW,EFFPWF,EFFTUR,EFFPSW,PWRTUR,CMAX,SYSEFF,CC
     2REJ,CFWPSW,QS,CQTUBE,NET,CYCEFF
      COMMON /CYCLE/ H1,H2,H4,H5,H5S,T1,T2
      COMMON /MISC/ MSG1,MSG2,MSG3,MSG4,MSG5,MSG6,MSG7
      COMMON /COST/ $BLR,$FDPMP,$ECIRC,$COND,$CCIRC,$TOT
      COMMON /MISCEL/ HT
      COMMON /SHELL/ AREAFR
      REAL *8MSG1,MSG2,MSG3,MSG4(2),MSG5(2),MSG6(2),MSG7(2),STRG
      REAL *8KC,NTUC,NTUB,NU,NETPWR,NET,NETOUT
C
C
      READ (5,20) MSG1,MSG2,MSG3
C
      READ (5,30) CLEAR,CCLR,THICK,CTHK,BTSWIN,CTSWIN,PCRIT,BKMET,CKMET
C
      READ (5,40) NETPWR
C
      READ (5,50) EFFPWF,EFFPSW,EFFTUR
C
      READ (5,60) MSG4,MSG5,MSG6,MSG7
C
      READ (5,70) FMBLR,FMCOND,FMFD,FMCIRC,DEXBLR,DEXCND,DEXFD,DEXCRC
      READ (5,30) HT,CR4,R4
      READ (5,40) AREAFR
C
```

```
      IF (N.EQ.11) READ (5,40) S,CS                                    READ 490
C                                                                      READ 500
C                                                                      READ 510
C     DETERMINE THE PROPERTIES OF THE WARM AND COLD SEAWATER           READ 520
C                                                                      READ 530
      T1 = BTSWIN                                                      READ 540
      T2 = CTSWIN                                                      READ 550
      VISSWH = 7.9554-.15215*T1+.0013673*T1**2-4.7842E-06*T1**3        READ 560
      VISSWC = 7.9554-.15215*T2+.0013673*T2**2-4.7842E-06*T2**3        READ 570
      CKSWH = .30927+5.69E-04*T1                                       READ 580
      CKSWC = .30927+5.69E-04*T2                                       READ 590
      CPSWH = 1.02645-6.86411E-04*T1+4.0374E-06*T1**2                  READ 600
      CPSWC = 1.02645-6.86411E-04*T2+4.0374E-06*T2**2                  READ 610
      RHOSWH = 62.39+.0036211*T1-7.625E-05*T1**2                       READ 620
      RHOSWC = 62.39+.0036211*T2-7.625E-05*T2**2                       READ 630
      PRSWH = 26.173-.54646*T1+.005147*T1**2-1.861E-05*T1**3           READ 640
      PRSWC = 26.173-.54646*T2+.005147*T2**2-1.861E-05*T2**3           READ 650
C                                                                      READ 660
      RETURN                                                           READ 670
C                                                                      READ 680
C                                                                      READ 690
C                                                                      READ 700
   20 FORMAT (3A8)                                                     READ 710
   30 FORMAT (8F10.0)                                                  READ 720
   40 FORMAT (3F10.0)                                                  READ 730
   50 FORMAT (3F2.2)                                                   READ 740
   60 FORMAT (8A8)                                                     READ 750
   70 FORMAT (8F3.2)                                                   READ 760
      END                                                              READ 770
```

```
      SLEROLTINE SETV
C
C     THIS SUBRCLTINE  SETS THE VARIABLES TC THE FRCPER VALUES
C
      IMPLICIT REAL*8(A-H,O-Z,$)
      COMMON /SHARE/ X(100),DEL(100),AX(100,100),N,M,MN,NP1,NM1
      COMMON /DIMEN/ VL,SA,HSC,HMAX,PITCH,CD,CI,CLEAR,DS,S,TUBEL,HS,HEIG
     1HT,AAVG,ATCT,ATOTC,BANKHT,HTMAX,ATCTB,THICK,CN,COC,CIC,CTLBL,CRITC
     2H,CCS,CCLR,CTHK,CS,NUP,NDN,NMAX,INDEX,NI,NIC
      COMMON /FLCW/ FLOWF,FLOSW,USW,RESW,UINF,UMAX,REAVG,FLCSWT,GPMWF,GP
     1MSW,CFLCSW,CUSW,CRESW,CGPMSW,CFLSWT
      COMMON /PRESS/ PC,P,R,BKE,KC,RHC,F,DELTAF,HEAD,HEADWF,DELPWF,CKC,C
     1F,CDELF,CHC,CRHO,CKE,FCOND
      REAL *8KC,NTUC,NTUB,NU,NETPWR,NET,NETCUT
C
C     THE FACTORS MULTIPLYING THE X'S ARE THE SCALING FACTORS.
C
      LSW = X(1)
      CLSW = X(2)
      PC = X(3)*100.
      FCOND = X(4)*100.
      TUBEL = X(5)*10.
      CTUBEL = X(6)*10.
      FLOWF = X(7)*1.E08
      DS = X(8)
      CDS = X(9)
      CD = X(10)
      COD = X(11)
C
C IF THE NUMBER OF SHELLS IS TO OPTIMIZED, THEN SET THOSE
C     VARIABLES ALSO.
C
      IF (N.EQ.11) GO TO 20
      S = X(12)
      CS = X(13)
   20 RETURN
      END
```

```
      SUBROUTINE GRAD1 (I)
C
C     THIS SUBROUTINE CALL ON SUMT TO FIND THE FIRST DERIVATIVE
C     NUMERICALLY.
C
      IMPLICIT REAL*8(A-H,O-Z,$)
      COMMON /SHARE/ X(100),DEL(100),A(100,100),N,M,MN,NP1,NM1
      REAL *8KC,NTUC,NTLB,NU,NETPWR,NET,NETCUT
      CALL DIFF1 (I)
      RETURN
      END
```

```
      SUBROUTINE MATRIX (J,L)
C
C     THIS SUBROUTINE CALL ON SUMT TO FIND THE SECOND DERIVATIVE
C     NUMERICALLY.
C
      IMPLICIT REAL*8(A-H,O-Z,$)
      COMMON /SHARE/ X(100),DEL(100),A(100,100),N,M,MN,NP1,NM1
      REAL *8KC,NTUC,NTLB,NU,NETPWR,NET,NETCUT
      CALL DIFF2 (J)
      RETURN
      END
```

```
      SUBROUTINE SETA (DI,TUBEL,PRSW)
C
C     THIS SUBROUTINE SETS THE  CONSTANTS  FOR THE COMPUTERS ROUTINE
C     'SIMQ' TO FIND THE COEFFICIENTS TO CURVE FIT  THE NUSSELT
C     NUMBER THROUGH THE TRANSISTION REGION (2000<RE<10000).
C
      IMPLICIT REAL*8(A-H,N-Z,$)
      COMMON /SIM/ A(16),B(4)
C
C     THE RIGHT HAND SIDE VECTOR
C
      B(1) = 1.
      B(2) = 1.
      B(3) = 0.
      B(4) = 0.
C
C     SET THE VALUES OF THE A MATRIX
C
      AA = .023*PRSW**.3
      R = 2000.
      CALL CLAM (R,NUL,DNUL,DI,TUBEL,PRSW)
      A1 = NUL-AA*R**.8
      B1 = DNUL-.8*AA*R**(-.2)
      AB = A1/B1
      A(1) = 8.E09
      A(2) = 1.2E07*AB+A(1)
      A(5) = 4.E06
      A(6) = 4.E03*AB+A(5)
      A(9) = R
      A(10) = AB+R
      A(13) = 1.
      A(14) = 1.
      R = 10000.
      CALL CLAM (R,NUL,DNUL,DI,TUBEL,PRSW)
      A2 = NUL-AA*R**.8
      B2 = DNUL-.8*AA*R**(-.2)
      AB = A2/B2
      A(3) = 1.E12
      A(4) = 3.E08*AB+A(3)
      A(7) = 1.E08
      A(8) = 2.E04*AB+A(7)
      A(11) = R
      A(12) = AB+R
      A(15) = 1.
      A(16) = 1.
      RETURN
      END
```

```
      SUBROUTINE CLAM (R,NUL,DNUL,DI,TUBEL,PRSW)                          CLAM  10
C                                                                         CLAM  20
C     THIS SUBROUTINE CALCULATES THE NUSSELT NUMBER AND THE PARTIAL OF    CLAM  30
C     THE NUSSELT NUMBER W.R.T. THE REYNOLDS NUMBER IN THE LAMINAR REGION CLAM  40
C     OF FLOW.                                                            CLAM  50
C                                                                         CLAM  60
      IMPLICIT REAL*8(A-H,N-Z,$)                                          CLAM  70
      A = 2.02*(PRSW*DI/(TUBEL*12.))**.333333                             CLAM  80
      B = A/3.                                                            CLAM  90
      NUL = A*R**.333333                                                  CLAM 100
      DNUL = B*R**(-.666667)                                              CLAM 110
C                                                                         CLAM 120
C     IF THE NUSSELT NUMBER IS LESS THAN 3.66 SET IT TO 3.66 AND THE      CLAM 130
C     PARTIAL TO 0.0                                                      CLAM 140
C                                                                         CLAM 150
      IF (3.66.LE.(A*12.59921)) GO TO 20                                  CLAM 160
      NUL = 3.66                                                          CLAM 170
      DNUL = 0.0                                                          CLAM 180
   20 RETURN                                                              CLAM 190
      END                                                                 CLAM 200
```

```
      SUBROUTINE AREA (II,MM)
C
C     THIS SUBROUTINE CALCULATES THE NUMBER OF TUBES IN THE BOILER AND THE
C     CONDENSER.
C
      IMPLICIT REAL*8(A-H,O-Z,$)
      COMMON /DIMEN/ VL,SA,HSC,HMAX,PITCH,OD,DI,CLEAR,DS,S,TUBEL,HS,HEIG
     1HT,AAVG,ATOT,ATOTC,BANKHT,HTMAX,ATOTB,THICK,CN,COD,CID,CTUBL,CPITC
     2H,CDS,CCLR,CTHK,CS,NUP,NDN,NMAX,INDEX,NI,NIC
      COMMON /FLOW/ FLOWF,FLOSW,USW,RESW,UINF,UMAX,REAVG,FLOSWT,GPMWF,GP
     1MSW,CFLOSW,CUSW,CRESW,CGPMSW,CFLSWT
      COMMON /PRESS/ PC,P,R,BKE,KC,RHO,F,DELTAP,HEAD,HEADWF,DELPWF,CKC,C
     1F,CDELP,CHD,CRHO,CKE,PCOND
      COMMON /TUBEN/ BNI,CNI
      COMMON /SHELL/ AREAFR
      REAL *8KC,NTUC,NTUB,NU,NETPWR,NET,NETOUT
      SAT(HS,AA,BB,AS) = .25*AS**2*DARCOS(AA)-HS*DSQRT(BB)
      CALL SETV
      CPITCH = COD+CCLR
      PITCH = OD+CLEAR
C
C     FIND THE CLEARANCE AREA
C
      IF (II.EQ.2) GO TO 20
C
C     FIND THE NO. OF TUBES IN THE BOILER
C
      ATB = .8660254*PITCH**2
      HS = .1*DS
      AA = 2.*HS/DS
      BB = .25*DS**2-HS**2
      SA = SAT(HS,AA,BB,DS)
      AS = DS-AREAFR
      AAP = 2.*HS/AS
      BBP = .25*AS**2-HS**2
      ASB = .78539816*AS**2-SAT(HS,AAP,BBP,AS)
      BNI = 144.*ASB/ATB
C
      RETURN
C
C     FIND THE NO. OF TUBES IN THE CONDENSER
C
   20 CAS = CDS-AREAFR
      ASC = .78539816*CAS**2
      ATC = .8660254*CPITCH**2
      CNI = 144.*ASC/ATC
      RETURN
      END
```

```
      SUBROUTINE OUTA
C
C  THIS SUBROUTINE PRINTS THE INFORMATION ABOUT THE SYSTEM
C
      IMPLICIT REAL*8(A-H,O-Z,$)
      COMMON /SHARE/ X(100),DEL(100),AX(100,100),N,M,MN,NP1,NM1
      COMMON /DIMEN/ VL,SA,HSC,HMAX,PITCH,OD,DI,CLEAR,DS,S,TUBEL,HS,HEIG
     1HT,AAVG,ATOT,ATOTC,BANKHT,HTMAX,ATOTB,THICK,CN,COD,CID,CTUBL,CPITC
     2H,CDS,CCLR,CTHK,CS,NUP,NDN,NMAX,INDEX,NI,NIC
      COMMON /FLOW/ FLOWF,FLOSW,USW,RESW,UINF,UMAX,REAVG,FLOSWT,GPMWF,GPC
     1MSW,CFLOSW,CUSW,CRESW,CGPMSW,CFLSWT
      COMMON /RESIST/ R1,R2,R4,R3C,R3B,HC,HCSW,HB,FACT,CWF,CSW,CMAX,CMIN
     1,UAC,NTUC,UAB,NTUB,RAT,GAMMA,EPS,CR1,CR2,CR3,CR4,CHFG,CHCSW,CUA,CN
     2TU,CCSW,CHSW
      COMMON /PROP/ RHOSWF,RHOSWC,VISSWF,VISSWC,CKSWF,CKSWC,PRSWH,PRSWC,
     1CPSWF,CPSWC,CKMET,BKMET,PCRIT,FMBLR,FMCIRC,FMFD,FMCOND,DEXBLR,DEXC
     2RC,DEXFD,DEXCND
      COMMON /PRESS/ PO,P,R,BKE,KC,RHC,F,DELTAP,HEAD,HEADWF,DELFWF,CKC,C
     1F,CDELP,CHD,CRHC,CKE,FCCND
      COMMON /TEMPER/ DELTOC,BTSWIN,TSWOC,DELTCB,TSWOB,DELT,TWALL,CTSWIN
     1,CTWOUT,CDELT
      COMMON /HEAT/ QTUBEC,QTUBEB,QSTUBB,CTEMP,CTTUBE,CTOT,CCTOT,CSTUBC,
     1QREJ,NETPWR,PWPWF,PWPSW,EFFPWF,EFFTUR,EFFPSW,PWRTUR,CMAX,SYSEFF,CQ
     2REJ,CPWPSW,CS,CCTUBE,NET,CYCEFF
      COMMON /COST/ $BLR,$FDPMP,$BCIRC,$CCND,$CCIRC,$TOT
      COMMON /CYCLE/ H1,H2,H4,H5,H5S,T1,T2
      COMMON /MISC/ MSG1,MSG2,MSG3,MSG4,MSG5,MSG6,MSG7
      COMMON /SIM/ A(16),B(4)
      COMMON /MISCEL/ HT
      COMMON /TUBEN/ BNI,CNI
      COMMON /MISC1/ N1,N2,N3
      COMMON /SSS/ HBA,CHCA
      REAL *8MSG1,MSG2,MSG3,MSG4(2),MSG5(2),MSG6(2),MSG7(2),STRG
      REAL *8KC,NTUC,NTUB,NU,NETPWR,NET,NETOUT
      DATA STRG/'********'/
C
C
      CALL RESTNT (0,VAL)
      $TOT = VAL*1.E07
      CALL RESTNT (N1,VAL)
      CALL RESTNT (N2,VAL)
      CALL RESTNT (N3,VAL)
      CHD = CDELP*144./RHOSWC
      HEAD = DELTAP*144./RHOSWH
      CMAX = 803.*PCRIT*(R**.35)*((1.-R)**.9)
      CMAX = CMAX/3.
      CSTUBB = 3.81972*CTUBEB/OD/TUBEL
      NET = PWRTUR-PWPWF
```

```
      CYCEFF = NET/QTOT
      NETOUT = PWRTUR-PWPWF-PWPSW-CPWPSW
      SYSEFF = NETOUT/QTOT
      CCSTUB = 3.819719*CCTUBE/(CCD*CTUBEL)
      VL = 276486.*RHOG(PC)*(SURFT(PC)/(RHOF(PC)-RHOG(PC)))**.5
      SAF = FLOWF/(VL*TUBEL*S)
C
C
C.............................................................................
C
C
C PRINT THE INFO ON THE PUMPS, THE COST DATA,AND THE POWER FLOWS
C
      WRITE (6,50) (STRG,I=1,12)
      WRITE (6,60)
      WRITE (6,70)
      WRITE (6,80)
      WRITE (6,90)
      WRITE (6,100) MSG5
      WRITE (6,110) T1,H1
      WRITE (6,120) DELTAP,HEAD
      WRITE (6,130) H2
      WRITE (6,140) GPMSW
      WRITE (6,150) T2,H4
      WRITE (6,160) EFFPSW
      WRITE (6,170) T1,H5
      WRITE (6,180)
      WRITE (6,190)
      WRITE (6,200) QTOT
      WRITE (6,210)
      WRITE (6,220) CQTOT
      WRITE (6,230) MSG5
      WRITE (6,240) CDELP,CHD
      WRITE (6,250) PWRTUR
      WRITE (6,260) CGPMSW
      WRITE (6,270) EFFPSW
      WRITE (6,280) PWPWF
      WRITE (6,290) PWPSW
      WRITE (6,300)
      WRITE (6,310) CPWPSW
      WRITE (6,320)
      WRITE (6,330) QREJ
      WRITE (6,340) MSG4
      WRITE (6,350) CQREJ
      WRITE (6,360) HEADWF,DELPWF
      WRITE (6,370) GPMWF
      WRITE (6,380) NETPWR
      WRITE (6,390) EFFPWF
```

```
      WRITE (6,400) NETCUT                                             OUTA 970
      WRITE (6,410) CYCEFF                                             OUTA 980
      WRITE (6,420) SYSEFF                                             OUTA 990
      WRITE (6,430) EFFTUR                                             OUTA1000
      WRITE (6,440)                                                    OUTA1010
      WRITE (6,450) $BLR,MSG6                                          OUTA1020
      WRITE (6,460) $FDPMF,MSG4                                        OUTA1030
      WRITE (6,470) $BCIRC,MSG5                                        OUTA1040
      WRITE (6,480) $CCIRC,MSG5                                        OUTA1050
      WRITE (6,490) $CCND,MSG7                                         OUTA1060
      WRITE (6,500) $TOT                                               OUTA1070
C                                                                      OUTA1080
C PRINT THE INFO ON THE BOILER                                         OUTA1090
C                                                                      OUTA1100
      WRITE (6,510) (STRG,I=1,8)                                       OUTA1110
      WRITE (6,520)                                                    OUTA1120
      WRITE (6,530)                                                    OUTA1130
      WRITE (6,540)                                                    OUTA1140
      WRITE (6,1130) RHCSWH                                            OUTA1150
      WRITE (6,550) DELTCB                                             OUTA1160
      WRITE (6,1150) VISSWH                                            OUTA1170
      WRITE (6,560) BTSWIN                                             OUTA1180
      WRITE (6,1170) CKSWH                                             OUTA1190
      WRITE (6,570) TSWCB                                              OUTA1200
      WRITE (6,1190) CPSWH                                             OUTA1210
      WRITE (6,1200) PRSWH                                             OUTA1220
      WRITE (6,580)                                                    OUTA1230
      WRITE (6,590)                                                    OUTA1240
      WRITE (6,600) MSG3                                               OUTA1250
      WRITE (6,610) CKMET                                              OUTA1260
      WRITE (6,620) R1,LAB                                             OUTA1270
      WRITE (6,630) R2,NTUB                                            OUTA1280
      WRITE (6,640)                                                    OUTA1290
      WRITE (6,660) R3B                                                OUTA1300
      WRITE (6,650) MSG2                                               OUTA1310
      WRITE (6,680) R4                                                 OUTA1320
      WRITE (6,670)                                                    OUTA1330
      WRITE (6,690) PCRIT                                              OUTA1340
      WRITE (6,700)                                                    OUTA1350
      WRITE (6,710)                                                    OUTA1360
      WRITE (6,720) HCSW                                               OUTA1370
      WRITE (6,730) DI,CD                                              OUTA1380
      WRITE (6,750) HB                                                 OUTA1390
      WRITE (6,740) TUBEL,PITCH                                        OUTA1400
      WRITE (6,20) HBA                                                 OUTA1410
      WRITE (6,760) CLEAR                                              OUTA1420
      WRITE (6,770) S                                                  OUTA1430
      WRITE (6,780)                                                    OUTA1440
```

```
      WRITE (6,750) DS
      WRITE (6,800) HEIGHT,MSG1
      WRITE (6,810) CSW
      WRITE (6,820)
      WRITE (6,830) BNI
      WRITE (6,840)
      WRITE (6,850) HS
      WRITE (6,860) FACT
      WRITE (6,870) DELT
      WRITE (6,890)
      WRITE (6,880)
      WRITE (6,900) QTUBEB
      WRITE (6,910) USW
      WRITE (6,940)
      WRITE (6,920)
      WRITE (6,950) FLOSWT
      WRITE (6,930) QSTUBB
      WRITE (6,970) FLOWF
      WRITE (6,960) QMAX
      WRITE (6,980)
      WRITE (6,1000) RESW
      WRITE (6,990)
      WRITE (6,1440) KC
      WRITE (6,1010) ATOTB
      WRITE (6,1450) BKE
      WRITE (6,1460) F
      WRITE (6,1020)
      WRITE (6,1030) VL
      WRITE (6,1040) SAF
      WRITE (6,30) SA
      WRITE (6,1050)
      WRITE (6,1060) PC
      WRITE (6,1070) P
C
C PRINT THE INFO ON THE CONDENSER
C
      WRITE (6,1080) (STRG,I=1,8)
      WRITE (6,1090)
      WRITE (6,1100)
      WRITE (6,1110)
      WRITE (6,1120) CR1,CUA
      WRITE (6,1130) RHOSWC
      WRITE (6,1140) CR2,CNTU
      WRITE (6,1150) VISSWC
      WRITE (6,1160) CR3
      WRITE (6,1170) CKSWC
      WRITE (6,1180) CR4
      WRITE (6,1190) CFSWC
```

AD-A031 709 NAVAL POSTGRADUATE SCHOOL MONTEREY CALIF F/G 10/2

AN OPTIMIZATION STUDY OF A LOW THERMAL POTENTIAL POWER SYSTEM.(U)

SEP 76 J R BUCKINGHAM, W M RAIKE

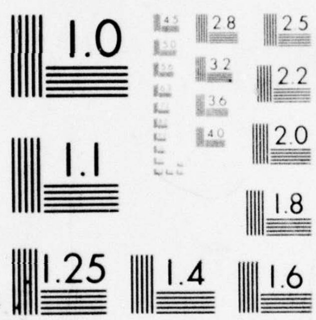
UNCLASSIFIED NPS-69KK76091 NL

2 OF 2

AD A031709

END

DATE FILMED 12-76



MICROCOPY RESOLUTION TEST CHART
NATIONAL BUREAU OF STANDARDS-1963-A

```
      WRITE (6,1200) PRSWC                                              OUTA1530
      WRITE (6,1210)                                                    OUTA1540
      WRITE (6,1220) MSG7                                               OUTA1550
      WRITE (6,1230) CHSW                                               OUTA1560
      WRITE (6,1240) CKMET                                              OUTA1570
      WRITE (6,1250) CHCSW                                              OUTA1580
      WRITE (6,40) CHCA                                                 OUTA1590
      WRITE (6,1260)                                                    OUTA2000
      WRITE (6,1270)                                                    OUTA2010
      WRITE (6,1280) CID,COD                                            OUTA2020
      WRITE (6,1290) CCTLBE                                             OUTA2030
      WRITE (6,1300) CTLBL,CPITCH                                       OUTA2040
      WRITE (6,1310) CCSTLB                                             OUTA2050
      WRITE (6,1320) CCLR                                               OUTA2060
      WRITE (6,1330) CS                                                 OUTA2070
      WRITE (6,1340) CDS                                                OUTA2080
      WRITE (6,1350)                                                    OUTA2090
      WRITE (6,830) CNI                                                 OUTA2100
      WRITE (6,1360)                                                    OUTA2110
      WRITE (6,1370) ATDTC                                              OUTA2120
      WRITE (6,1380) CUSW                                               OUTA2130
      WRITE (6,1390)                                                    OUTA2140
      WRITE (6,1400) CFLSWT                                             OUTA2150
      WRITE (6,1410) FLOWF                                              OUTA2160
      WRITE (6,1420)                                                    OUTA2170
      WRITE (6,1430) CRESW                                              OUTA2180
      WRITE (6,1440) CKC                                                OUTA2190
      WRITE (6,1450) CKE                                                OUTA2200
      WRITE (6,1460) CF                                                 OUTA2210
      WRITE (6,1470)                                                    OUTA2220
      WRITE (6,1480) PCCND                                              OUTA2230
      WRITE (6,1490)                                                    OUTA2240
      WRITE (6,1500) CDELT                                              OUTA2250
      WRITE (6,1510) CTSWIN                                             OUTA2260
      WRITE (6,1520) CTWCLT                                             OUTA2270
      RETURN                                                            OUTA2280
C                                                                       OUTA2290
C                                                                       OUTA2300
   20 FORMAT ('+',T67,'ASSUMED VALUE =',F7.2)                           OUTA2310
   30 FORMAT (' ',T16,'ACTUAL=',F10.3,' (FT2)')                         OUTA2320
   40 FORMAT (' ',T63,'ASSUMED VALUE=',F9.2)                            OUTA2330
   50 FORMAT ('1',T21,'SYSTEM DATA',T87,'PUMP DATA',/,'+',2A8,'---------OUTA2340
     1',6A8,'                ',4A8)                                     OUTA2350
   60 FORMAT ('0',       'CYCLE STATE POINTS',/,'+',      '-----------------OUTA2360
     1')                                                                OUTA2370
   70 FORMAT ('+',T63,'CIRC PUMP (BOILER)')                             OUTA2380
   80 FORMAT (' ',T24,'TEMPERATURE(F)    ENTHALPY(BTU/LBM)')            OUTA2390
   90 FORMAT ('+',T65,'TYPE: AXIAL FLOW')                               OUTA2400
```

```
100 FORMAT (' ',T65,'MATERIAL: ',2A8)                                       OUTA2410
110 FORMAT (' ',T02,'CONDENSER EXIT (1)',7X,F7.3,10X,F8.3)                  OUTA2420
120 FORMAT ('+',T65,'PRESSURE DROP=',F9.6,' (PSIA)  OR ',F9.6,' (FT)')OUTA2430
130 FORMAT (' ',T07,'PUMP EXIT (2)',24X,F8.3)                               OUTA2440
140 FORMAT ('+',T65,'FLOW RATE PER SHELL=',F9.2,' (GPM)')                   OUTA2450
150 FORMAT (' ',T05,'BOILER EXIT (3)',7X,F7.3,10X,F8.3)                     OUTA2460
160 FORMAT ('+',T65,'EFFICIENCY=',F6.4,' (ASSUMED)')                        OUTA2470
170 FORMAT (' ',T04,'TURBINE EXIT (4)',7X,F7.3,10X,F8.3)                    OUTA2480
180 FORMAT ('0',     'SYSTEM POWER BUDGET'/'+',     '__________________OUTA2490
   1')                                                                      OUTA2500
190 FORMAT ('+',T63,'CIRC PUMP (CONDENSER)')                                OUTA2510
200 FORMAT ('0',T03,'REQUIRED HEAT ABSORPTION=',E16.7,' (BTU/HR)')          OUTA2520
210 FORMAT ('+',T65,'TYPE: AXIAL FLOW')                                     OUTA2530
220 FORMAT (' ',T10,'ACTUAL ABSORPTION=',E16.7,' (BTU/HR)')                 OUTA2540
230 FORMAT ('+',T65,'MATERIAL: ',2A8)                                       OUTA2550
240 FORMAT (' ',T65,'PRESSURE DROP=',F9.6,' (PSIA)  OR ',F9.6,' (FT)')OUTA2560
250 FORMAT (' ',T10,'TURBINE POWER OUT=',E16.7,' (BTU/HR)')                 OUTA2570
260 FORMAT ('+',T65,'FLOW RATE PER SHELL=',F9.2,' (GPM)')                   OUTA2580
270 FORMAT (' ',T65,'EFFICIENCY=',F6.4,' (ASSUMED)')                        OUTA2590
280 FORMAT (' ',T12,'FEED PUMP POWER=',E16.7,' (BTU/HR)')                   OUTA2600
290 FORMAT ('0',T06,'CIRC PUMP POWER (BLR)=',E16.7,' (BTU/HR)')             OUTA2610
300 FORMAT (' ',T63,'WORKING FLUID PUMP')                                   OUTA2620
310 FORMAT (' ',T05,'CIRC PUMP POWER (COND)=',E16.7,' (BTU/HR)')            OUTA2630
320 FORMAT (' ',T65,'TYPE: CENTRIFUGAL')                                    OUTA2640
330 FORMAT (' ',T05,'REQUIRED HEAT REJECTED=',E16.7,' (BTU/HR)')            OUTA2650
340 FORMAT ('+',T65,'MATERIAL: ',2A8)                                       OUTA2660
350 FORMAT (' ',T12,'ACTUAL REJECTED=',E16.7,' (BTU/HR)')                   OUTA2670
360 FORMAT ('+',T65,'TOTAL HEAD=',F8.3,' (FT OF FLUID)',3X,'OR',3X,F8.OUTA2680
   12,' (PSIA)')                                                            OUTA2690
370 FORMAT (' ',T65,'FLOW RATE PER SHELL=',F9.2,' (GPM)')                   OUTA2700
380 FORMAT (' ',T09,'REQUIRED NET POWER=',E16.7,' (BTU/HR)')                OUTA2710
390 FORMAT ('+',T65,'EFFICIENCY=',F6.4,' (ASSUMED)')                        OUTA2720
400 FORMAT (' ',T15,'ACTUAL POWER=',E16.7,' (BTU/HR)')                      OUTA2730
410 FORMAT ('0',T09,'   CYCLE EFFICIENCY=',F7.5)                            OUTA2740
420 FORMAT (' ',T10,'SYSTEM EFFICIENCY=',F7.5)                              OUTA2750
430 FORMAT ('0',T09,'TURBINE EFFICIENCY=',F4.2,' (ASSUMED)')                OUTA2760
440 FORMAT ('0',T02,'COST DATA'/'+',T02,'_____________****************OUTA2770
   1****************************')                                          OUTA2780
450 FORMAT ('0',T03,'BOILER    =$',F9.0,3X,'MTL: ',2A8)                     OUTA2790
460 FORMAT (' ',T03,'FEED PUMP=$',F9.0,3X,'MTL: ',2A8)                      OUTA2800
470 FORMAT (' ',T03,'BLR CIRC =$',F9.0,3X,'MTL: ',2A8)                      OUTA2810
480 FORMAT (' ',T03,'COND CIRC=$',F9.0,3X,'MTL: ',2A8)                      OUTA2820
490 FORMAT (' ',T03,'CONDENSER=$',F9.0,3X,'MTL: ',2A8)                      OUTA2830
500 FORMAT (' ',T06,'TOTAL=$',F10.0)                                        OUTA2840
510 FORMAT ('1',T40,'BOILER DATA',/,'+',4A8,'______________________',OUTA2850
   1 4A8)                                                                   OUTA2860
520 FORMAT ('0','PROPERTIES'/'+','__________')                              OUTA2870
530 FORMAT ('+',T62,'TEMPERATURES',/,'+',T62,'____________')                OUTA2880
```

```
540 FORMAT ('0 SALTWATER')                                              OUTA2890
550 FORMAT ('+',T63,'DELTA TO BOILING =',F8.4,' (F)')                   OUTA2900
560 FORMAT ('+',T63,'SEAWATER INLET   =',F8.4,' (F)')                   OUTA2910
570 FORMAT ('+',T63,'SEAWATER OUTLET  =',F8.4,' (F)')                   OUTA2920
580 FORMAT ('0 TUBES')                                                  OUTA2930
590 FORMAT ('+',T62,'INTERMEDIATE CALCULATIONS',/,'+',T62,'__________   OUTA2940
   1_________')                                                         OUTA2950
600 FORMAT ('      MATERIAL: ',1A8)                                     OUTA2960
610 FORMAT ('      CONDUCTIVITY=',F10.5,' (BTU/HR-FT-F)')               OUTA2970
620 FORMAT ('+',T63,'R1 =',F12.10,'       UA  =',F9.2)                  OUTA2980
630 FORMAT (' ',T63,'R2 =',F12.10,'       NTU =',F9.6)                  OUTA2990
640 FORMAT ('   WORKING FLUID')                                         OUTA3000
650 FORMAT ('      MATERIAL: ',1A8)                                     OUTA3010
660 FORMAT ('+',T63,'R3 =',F12.10)                                      OUTA3020
670 FORMAT ('      PROPERITES ARE PRESS AND TEMP DEP')                  OUTA3030
680 FORMAT ('+',T63,'R4 =',F12.10)                                      OUTA3040
690 FORMAT ('      P CRITICAL=',F7.2,' (PSIA)')                         OUTA3050
700 FORMAT (' ',T62,'HEAT TRANSFER COEFFICIENTS'/'+',T62,'____________ OUTA3060
   1__________')                                                        OUTA3070
710 FORMAT (' DIMENSIONS'/'+__________')                                OUTA3080
720 FORMAT (' ',T63,'WATERSIDE =',F7.2)                                 OUTA3090
730 FORMAT ('   TUBE ID=',F7.4,' (IN)      TUBE OD=',F7.4,' (IN)')      OUTA3100
740 FORMAT ('   TUBE LENGTH=',F7.3,' (FT) TUBE PITCH=',F6.3,' (IN)')    OUTA3110
750 FORMAT ('+',T63,'WORKING FLUID SIDE =',F7.2)                        OUTA3120
760 FORMAT ('   TUBE CLEARANCE=',F6.3,' (IN)')                          OUTA3130
770 FORMAT ('0 NO. OF SHELLS=',F7.2)                                    OUTA3140
780 FORMAT ('+',T62,'CAPACITY RATE (FLOW X C/P)'/'+',T62,'____________ OUTA3150
   1')                                                                  OUTA3160
790 FORMAT ('   SHELL DIAMETER=',F7.3,' (FT)')                          OUTA3170
800 FORMAT ('   TUBE BANK HEIGHT=',F7.3,' (FT) BASED ON ',1A8)          OUTA3180
810 FORMAT ('+',T63,'SALTWATER      =',F10.3)                           OUTA3190
820 FORMAT (' ')                                                        OUTA3200
830 FORMAT ('   TUBES PER SHELL: REQ #=',F9.2)                          OUTA3210
840 FORMAT (' ',T62,'HEAT FLOWS'/'+',T62,'__________')                  OUTA3220
850 FORMAT ('   TUBE BANK HEIGHT ABOVE CENTERLINE=',F7.4,' (FT)')       OUTA3230
860 FORMAT (' ',T63,'BUNDLE FACTOR=',F6.3)                              OUTA3240
870 FORMAT (' ',T66,'FOR (TB - TSAT)=',F8.3,' (F)')                     OUTA3250
880 FORMAT ('+',T63,'PER TUBE')                                         OUTA3260
890 FORMAT (' FLOW CALCULATIONS'/'+_________________')                  OUTA3270
900 FORMAT (' ',T66,'BOILING=',F9.2,' (BTU/HR)')                        OUTA3280
910 FORMAT ('   SALTWATER VELOCITY=',F8.4,' (FT/SEC)')                  OUTA3290
920 FORMAT ('+',T63,'PER UNIT AREA')                                    OUTA3300
930 FORMAT ('+',T66,'BOILING=',F9.2,' (BTU/HR-FT2)')                    OUTA3310
940 FORMAT ('   MASS FLOW RATE THRU THE BOILER')                        OUTA3320
950 FORMAT ('      SALTWATER     = ',E16.8,' LBM/HR)')                  OUTA3330
960 FORMAT ('+',T66,'MAXIMUM=',F9.2 ,' (BTU/HR-FT2)')                   OUTA3340
970 FORMAT ('      WORKING FLUID= ',E16.8,' (LBM/HR)')                  OUTA3350
980 FORMAT ('   REYNOLDS NUMBERS')                                      OUTA3360
```

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 990 FORMAT ('+',T62,'HEAT EXCHANGER AREAS'/'+',T62,'___________________OUTA3370
    1__')                                                                OUTA3380
1000 FORMAT ('     SALTWATER      =',F10.1)                              OUTA3390
1010 FORMAT ('+',T63,'TOTAL PER SHELL (OD) =',E16.7,' (FT2)')            OUTA3400
1020 FORMAT (//'OMISCELLANEOUS DATA'/'+__________________')              OUTA3410
1030 FORMAT ('O ALLOWABLE VAPOR LOAD=',F9.2,' (LBM/HR-FT3)')             OUTA3420
1040 FORMAT ('   SEPARATION AREA REQ=',F10.3,' (FT2)')                   OUTA3430
1050 FORMAT (//'OPRESSURES'/'+_________')                                OUTA3440
1060 FORMAT ('O BOILER OUTLET =',F7.2,' (PSIA)')                         OUTA3450
1070 FORMAT ('   BUNDLE AVERAGE=',F7.2,' (PSIA)')                        OUTA3460
1080 FORMAT ('1',T37,'CONDENSER DATA',/,'+',4A8,'___________________',OUTA3470
    1 4A8)                                                               OUTA3480
1090 FORMAT ('0',     'MATERIALS',/,'+_________')                        OUTA3490
1100 FORMAT ('+',T62,'INTERMEDIATE CALCULATIONS',/,'+',T62,'__________ OUTA3500
    1_______________')                                                   OUTA3510
1110 FORMAT ('0',T03,'SALTWATER')                                        OUTA3520
1120 FORMAT ('+',T62,' R1=',F12.10,'      UA=',F9.2)                     OUTA3530
1130 FORMAT (' ',T05,'DENSITY          =',F5.2,' (LBM/FT3)')             OUTA3540
1140 FORMAT ('+',T62,' R2=',F12.10,'     NTU=',F9.6)                     OUTA3550
1150 FORMAT (' ',T05,'VISCOSITY        =',F6.3,' (LBM/HR-FT)')           OUTA3560
1160 FORMAT ('+',T62,' R3=',F12.10)                                      OUTA3570
1170 FORMAT (' ',T05,'CONDUCTIVITY =',F7.4,' (BTU/HR-FT-F)')             OUTA3580
1180 FORMAT ('+',T62,' R4=',F12.10)                                      OUTA3590
1190 FORMAT (' ',T05,'SPECIFIC HEAT=',F7.4,' (BTU/LBM-F)')               OUTA3600
1200 FORMAT (' ',T05,'PRANDTL NO.   =',F6.3)                             OUTA3610
1210 FORMAT ('+',T62,'HEAT TRANSFER COEFFICIENTS',/,'+',T62,'_________OUTA3620
    1_________________')                                                 OUTA3630
1220 FORMAT ('0',     ' TUBE: ',2A8)                                     OUTA3640
1230 FORMAT ('+',T62,' WATERSIDE      =',F9.2)                           OUTA3650
1240 FORMAT (' ',T04,' CONDUCTIVITY=',F7.3,' (BTU/HR-FT-F)')             OUTA3660
1250 FORMAT ('+',T62,' WORKING FLUID=',F9.2)                             OUTA3670
1260 FORMAT ('0',     'DIMENSIONS'/'+__________')                        OUTA3680
1270 FORMAT ('+',T62,'HEAT FLOWS',/,'+',T62,'__________')                OUTA3690
1280 FORMAT ('0',     ' TUBE ID=',F7.4,' (IN)        TUBE OD    =',F7.4,' (OUTA3700
    1IN)')                                                               OUTA3710
1290 FORMAT ('+',T62,' PER TUBE       =',F9.2,' (BTU/HR)')               OUTA3720
1300 FORMAT (' ',     ' TUBE LENGTH=',F7.3,' (FT)  TUBE PITCH=',F6.3,' (OUTA3730
    1IN)')                                                               OUTA3740
1310 FORMAT ('+',T62,' PER UNIT AREA=',F9.2,' (BTU/HR-FT2)')             OUTA3750
1320 FORMAT (' ',     ' TUBE CLEARANCE=',F6.3,' (IN)')                   OUTA3760
1330 FORMAT ('0',     ' NO. OF SHELLS=',F7.2)                            OUTA3770
1340 FORMAT (' ',     ' SHELL DIAMETER=',F7.3,' (FT)')                   OUTA3780
1350 FORMAT ('+',T62,'HEAT EXCHANGER AREA',/,'+',T62,'________________OUTA3790
    1 ')                                                                 OUTA3800
1360 FORMAT ('0',     'FLOW CALCULATIONS'/'+_________________')          OUTA3810
1370 FORMAT ('+',T62,' TOTAL PER SHELL (OD) =',E16.7,' (FT2)')           OUTA3820
1380 FORMAT ('0',     ' SALTWATER VELOCITY=',F8.4,' (FT/SEC)')           OUTA3830
1390 FORMAT ('0',     ' MASS FLOW RATES')                                OUTA3840
```

```
1400 FORMAT (' ',T02,'   SALTWATER     =',E17.8,' (LBM/HR)')          OUTA3850
1410 FORMAT (' ','   WORKING FLUID= ',E16.8,' (LBM/HR)')             OUTA3860
1420 FORMAT ('0',' REYNOLDS NUMBERS')                                 OUTA3870
1430 FORMAT (' ',T05,'   SALTWATER=',F10.1)                           OUTA3880
1440 FORMAT ('0',' TUBE CONTRACTION FACTOR=',F7.4)                    OUTA3890
1450 FORMAT (' ',' TUBE EXPANSION FACTOR   =',F7.4)                   OUTA3900
1460 FORMAT (' ',' FRICTION FACTOR(FANNING)=',F9.7)                   OUTA3910
1470 FORMAT ('0','PRESSURES'/'+________')                             OUTA3920
1480 FORMAT ('0',' CONDENSER AVG=',F7.2,' (PSIA)')                    OUTA3930
1490 FORMAT ('0','TEMPERATURES'/'+____________')                      OUTA3940
1500 FORMAT ('0',T03,'TSAT - TWALL    =',F8.4,' (FT)')                OUTA3950
1510 FORMAT (' ',T03,'SEAWATER INLET =',F8.4,' (F)')                  OUTA3960
1520 FORMAT (' ',' SEAWATER OUTLET=',F8.4,' (F)')                     OUTA3970
```

```
C<<<<<<<<<<<<<<<<<<<<<<<<<  PROPERTIES OF PROPANE  >>>>>>>>>>>>>>>>>>>>>>>>>

C  SATURATION TEMP
C
      FUNCTION TSAT (P)
      IMPLICIT REAL*8(A-H,O-Z,$)
      TSAT = -6.22162E01+2.08944E00*P-1.39456E-02*P**2+5.82940E-05*P**3-
     11.00051E-07*P**4
      RETURN
      END
C  ENTHALPY OF THE SATURATED LIQUID
C
      FUNCTION HF (P)
      IMPLICIT REAL*8(A-H,O-Z,$)
      HF = 8.51215E01+6.30546E-01*P-1.04803E-03*P**2
      RETURN
      END
C  ENTHALPY OF THE SATURATED VAPOR
C
      FUNCTION HG (P)
      IMPLICIT REAL*8(A-H,O-Z,$)
      HG = 2.69666E02+2.82932E-01*P-5.85830E-04*P**2
      RETURN
      END
C  LIQUID DENSITY
C
      FUNCTION RHOF (P)
      IMPLICIT REAL*8(A-H,O-Z,$)
      RHOF = 3.64927E01-6.12201E-02*P+1.95928E-04*P**2-3.57024E-07*P**3
      RETURN
      END
C  VAPOR DENSITY
C
      FUNCTION RHOG (P)
      IMPLICIT REAL*8(A-H,O-Z,$)
      RHOG = 4.21893E-02+8.79332E-03*P
      RETURN
      END
C  SURFACE TENSION
C
      FUNCTION SURFT (P)
      IMPLICIT REAL*8(A-H,O-Z,$)
      SURFT = 5.9381E-10*(RHOF(P)-RHOG(P))**4
      RETURN
      END
```

```
C HEAT CAPACITY CF THE LICUIC
C
      FLNCTICN CFF (T)
      IPFLICIT REAL*8(A-H,O-Z,$)
      CFF = 5.6202E-01+6.57786E-04*T+2.08525E-06*T**2
      RETLRN
      ENC
C  LICLIC'S VISCOSITY
C
      FLNCTICN VISCF (T)
      IPFLICIT REAL*8(A-H,O-Z,$)
      VISCF = (1.08866E01-6.0602E-02*T+2.60276E-04*T**2-1.18599E-06*T**3
     1)*1.E-05
      RETLRN
      ENC
C  VAPCR VISCCSITY
C
      FLNCTION VISCSV (T)
      IPFLICIT REAL*8(A-H,O-Z,$)
      VISCSV = (4.78332+1.20747E-02*T+2.62346E-05*T**2)*1.E-06
      RETLRN
      ENC
C  PRANCTL NUMBER OF PRCPANE
C
      FLNCTION PRWF (T)
      IPFLICIT REAL*8(A-H,O-Z,$)
      PRWF = CPF(T)*VISCF(T)/TKF(T)
      RETURN
      ENC
C  SATLRATED LICLID ENTRCPY
C
      FLNCTION ENTF (T)
      IPFLICIT REAL*8(A-H,O-Z,$)
      ENTF = 2.28435E-01+1.17812E-03*T+1.85814E-06*T**2
      RETLRN
      ENC
C  SATLRATED VAPCR ENTRCPY
C
      FLNCTION ENTG (T)
      IPFLICIT REAL*8(A-H,O-Z,$)
      ENTG = 6.0467E-01-2.2967E-04*T+1.0989E-06*T**2
      RETLRN
      ENC
```

```
C
C     THERMAL CONDUCTIVITY OF THE LIQUID
      FUNCTION TKF (T)
      IMPLICIT REAL*8(A-H,O-Z,$)
      TKF = .06235-.133E-03*T
      RETURN
      END
```

SAMPLE DATA

```
* PARAMETER CARD
1.      E-071.      E 001.      E-054.      E 001.8      E 030018001300003

* INITIAL X VECTOR
3.      D 003.      D 001.17    D 00.99     D 002.9      D 002.9      D 00TURBULEN
6.      D 002.      D 013.5     D 012.      D 002.       D 001.5      D 00TURBULEN
1.5     D 00

* DATA FOR THE PROBLEM
HMAX    PROPANE CU90-NI
.5      .5          .028        .028        75.          40.          661.5        35.5
35.5
85555COO.
555555
BRONZE            STAINLESS STEEL CS/BRASS          CS/BRASS
1521521281641C0100100100100
15.     0.0         0.0
.83333333

* FIRST OPTION CARD
     3       1       1       1       2       2       2               3       1

* TOLERANCE CARD
1.      E-040.0     E 00

* SECOND OPTION CARD
     1       4
```

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